Program 60-1163--Simple Epicyclic Differential Design

Introduction

The simple epicyclic gear unit consists of a central external gear (sun gear) meshed with one or more external gears (planet gears). The planet gears are then meshed with an internal gear (ring gear) which encloses the system. The planet gears and planet gear support bearings are held in a carrier which rotates about the geometric center of the unit. The term "epicyclic" comes from the path of a point on a planet gear which traces out an epicycloid in space. Therefore, there are three input/output elements in simple epicyclic gear differentials.

The ring/sun ratio range for which these units can be designed with reasonable proportions is about 2:1 to 11:1.

Below this range the planet gears become quite small and it becomes difficult to design the gears and planet bearings for reasonable life. Above this range the sun gear becomes small and the number of planets that can be used without interference is limited. This, again, makes the design of the bearings difficult. The range can be extended using compound epicyclic differentials. (See UTS TK model 60-1164, Compound Epicyclic Differentials.)

If more than one planet gear is used the number of planets that will assemble between the sun and ring is limited by the numbers of teeth in the sun and ring and by the possibility of interference between the tips of the planet gear teeth. For a number of planets to assemble equally spaced around the center, the sum of the tooth numbers in the ring and sun divided by the number of planets used must be an integer:

$$(N_{ring}+N_{sun})/n_p = \text{integer}$$

where: N_{ring} = Number of teeth in ring gear
N_{sun} = Number of teeth in sun gear
n_p = Number of planet gears

The distance between the planet gear centers in the carrier must, of course, be greater than the outside diameter of the planet gears or tooth tip interference will result (assuming the planet gears are in the same plane).

It is not necessary that the planets be equally spaced. However, to make assembly possible, they must be spaced at multiples of the "Least mesh angle".

$e_p/ß$ = integer

$ß$ = $360°/(N_{ring}+N_{sun})$

where: e_p = Angle between adjacent planet gears, deg
$ß$ = Least mesh angle, deg

For example, suppose we have an epicyclic set with N_{ring} = 68 teeth and N_{sun} = 18 teeth and we wish to use 4 planets arranged 90 degrees apart. $(N_{ring}+N_{sun})/4 = 21.5$ which is not an integer so we cannot arrange 4 planets 90 degrees apart. $(N_{ring}+N_{sun})/2 = 43$ which is an integer so we can arrange 2 planets 180 degrees apart. The least mesh angle, $ß = 360°/(N_{ring}+N_{sun}) = 4.186$ degrees. When we attempt to place a planet 90 degrees from the first planet we find that we are at $90°/ß = 21.5$ least mesh angles and cannot assemble. We can, however, place the planet at 21 or 22 least mesh angles. This would put the planet gear at 1/2 ß or 2.093 degrees from 90 degrees. Then, since we know that 2 planets will assemble 180 degrees apart, the 4 planets would be placed at 0 degrees, 87.907 degrees, 180 degrees and 267.907 degrees. The tip clearance should then be checked. Since we have two sets of planets 180 degrees apart the (theoretical) summation of the bearing loads on the sun and ring is still zero.

The model will calculate the planet interference outside diameter. The planet OD must be less than this diameter. The planet tooth tip clearance will be the amount the actual planet OD is less than the interference OD.

The summation of radial loads on the sun and ring will be made to determine whether or not the radial loads on the sun and planet are balanced. The result is displayed as a “Yes” or “No” under “Radial Loads On Sun & Ring Balanced?”.

It is not necessary (or even desirable) that $N_{ring} = N_{sun} + 2*N_{planet}$. If this relationship is met and the center distance is “standard” then the operating pressure angles at the sun/planet external mesh, ϕ_{ext}, and the planet/ring internal mesh, ϕ_{int}, will be equal to the nominal pressure angle of the system. If ϕ_{ext} is made higher than nominal and ϕ_{int} lower than nominal it will increase the strength of the set and reduce the burst stress on the ring. ϕ_{ext} and ϕ_{int} can be easily controlled by the number of planet teeth and the operating center distance. (If the ring gear rim thickness is 2 tooth depths or more, a high operating pressure angle will tend to reduce the bending stress. If the ring gear rim thickness is 1.5 tooth depths or less, a low operating pressure angle will tend to reduce the bending stress.)

The pitch line velocity is calculated and the minimum recommended AGMA quality class is determined is accordance with the ANSI/AGMA 6023-A88 Design Manual for Enclosed Epicyclic Gear Drives.

The percent factorization is also calculated in accordance with ANSI/AGMA 6023-A88. See the standard for more information on factorized tooth numbers. The fundamental meshing frequency is displayed along with the expected primary torsional variation frequency.

The model contains a table of K factors and Unit Loads for use in estimating the size of the gears required to carry the necessary load. The K factor is a function of the compressive stress carried on the teeth and is proportional to the square of the stress. The Unit Load is a function of the bending stress in the root area of the gears and is directly proportional to the stress. Both factors are directly proportional to the load. Different gear materials are, of course, capable of carrying different K and Unit Load factors for a given number of cycles. These factors are approximate because they do not contain many of the elements affecting the stresses on the gears. They are close enough, however, to allow us to get a "starting place" for our design with only the information at hand. It is essential, of course, to check our preliminary design with equations containing all the factors known to affect the operation and life of the gear set.

The selection of the K factor and Unit Load is, of course, based on the material used for the gears and our best estimate of the load the gears will carry. The number of cycles the gears are required to run will also be part of the selection process.

The first thing we need to determine is a "service factor" which adjusts the load to account for the extra load imposed on the gears from non-uniform torques produced by the driver and driven machines. A few selected service factors are contained in the table "SF". The number of cycles we must run will be dealt with separately from the service factor and the service factors listed do not include adjustments for duration of service. Further information on service factors can be found in various AGMA standards pertaining to specific industries and applications. The service factors usually applied sometimes are not sufficient for critical drives running at high power and/or speed and must be used with caution. AGMA Standard 427, "AGMA Information Sheet, Systems Considerations for Critical Service Gear Drives" is an excellent source for information concerning the rating of these drives.

INT TABLE: SF

Element	Prime Mover		Driven Machine	
1		Uniform	Mod Shock	Heavy Shock
3	Electric Motor	1.00	1.25	1.75
4	Turbine			
6	Hydraulic Motor	1.10	1.35	1.85
8	Multi-Cylinder	1.25	1.50	2.00
9	IC Engine			
11	Single-Cylinder	1.50	1.75	2.25
12	IC Engine			

The table “KUL” contains K factors and Unit Loads for a number of materials and conditions for steel gears.

INT TABLE: KUL

Element	Application	AGMA Q#	Type/Heat	*Pin Cy	*SF	*Ratio	K, psi	UL, psi
1	HI SPEED	Q12&up	HELICAL					
2			Carb	2E10	1.00	4.000	420	8000
3			BHN 360	2E10	1.00	4.000	240	5900
4			BHN 240	2E10	1.00	4.000	130	4600
5	MED SPEED	Q10-11	HELICAL					
6			Carb	1E9	1.00	4.000	550	8000
7			BHN 360	1E9	1.00	4.000	320	5900
8			BHN 240	1E9	1.00	4.000	170	4600
9		Q10-11	SPUR					
10			Carb	1E9	1.00	4.000	320	6200
11			BHN 360	1E9	1.00	4.000	190	4600
12			BHN 240	1E9	1.00	4.000	100	3600
13		Q8-9	HELICAL					
14			Carb	1E9	1.00	4.000	460	6600
15			BHN 360	1E9	1.00	4.000	260	4900
16			BHN 240	1E9	1.00	4.000	140	3800
17		Q8-9	SPUR					
18			Carb	1E9	1.00	4.000	270	5200
19			BHN 360	1E9	1.00	4.000	160	3800
20			BHN 240	1E9	1.00	4.000	80	3000
21	LOW SPEED	Q6-7	HELICAL					
22			Carb	1E8	1.00	4.000	510	5900
23			BHN 360	1E8	1.00	4.000	290	4300
24			BHN 240	1E8	1.00	4.000	150	3400
25		Q6-7	SPUR					
26			Carb	1E8	1.00	4.000	280	4500
27			BHN 360	1E8	1.00	4.000	160	3300
28			BHN 240	1E8	1.00	4.000	80	2600
29	AEROSPACE	Q12&up	HELICAL	1E9	1.00	5.000	1220	11500
30	Carb		SPUR	1E9	1.00	2.000	710	8400
31	VEHICLE							
32	Main-Carb	Q8-10	HELICAL	5E7	1.00	5.000	900	11400
33			SPUR	5E7	1.00	3.000	520	8700
34	Final-Carb	Q8-10	HELICAL	2E6	1.00	4.000	1890	11600
35			SPUR	2E6	1.00	3.000	1260	10600

The table takes into account the class of gearing, such as high speed or medium speed. The accuracy to which the gears are made is also included along with the type of gear (spur, helical) and the heat treatment used.

The unit loads have been adjusted for reverse bending of the planet gear teeth by reducing the unit loads usually used to 70% of the normal values. The table is "interactive" and we can change the items marked with an asterisk.

The number of cycles needed is found by multiplying the speed (relative to the carrier) by the life required. (If the sun meshes with more than one planet this must be taken into account.) You may move the cursor to the appropriate location in the table and change the values for pinion cycles, service factor and ratio to suit the application. After solving the model the table will be updated to reflect the changed data.

Examples

If you are using model 60-1163 for the first time, you may wish to run the following examples. (Some output values shown here have been rounded off.)

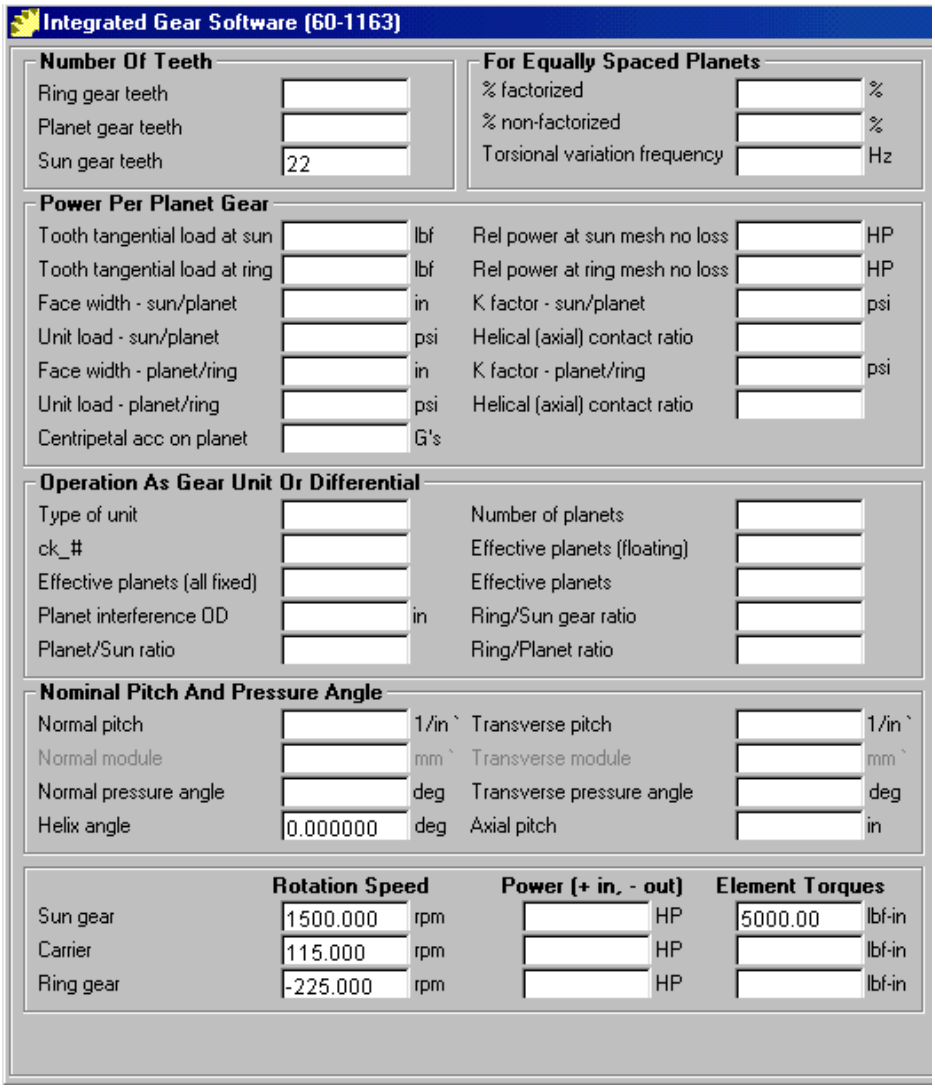
Example 1

Suppose we wish to design a spur gear differential planetary set with a sun gear speed of 1500 RPM, a ring gear speed of 225 RPM in the opposite direction from the sun, and a carrier speed of about 115 RPM in the same direction as the sun. The sun gear torque for this condition is 5000 lb-in. Also, the smallest number of teeth we wish to use is about 22. (The number of teeth would be selected based upon material and duty cycle; see UTS model 60-180.)

Figure 1-1 shows a portion of the 60-1163 wizard form with the initial conditions entered. (If the sun and carrier speeds are input as positive the ring must be input as negative to indicate opposite rotation.) Check and uncheck the "Enable table?" checkbox in the lower right of the form and press Enter to clear the interactive table. Report 1-1 shows the solved model.

UTS Integrated Gear Software

Fig. 1-1



Report 1-1

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
MESSAGE FIELD			
ERROR MESSAGE, internal mesh			
ERROR MESSAGE, external mesh			
ERROR MESSAGE, mesh - general			
Prime factors greater than 100	**unknown**		
NUMBER OF TEETH			
Ring Gear Teeth	**90**		
Planet Gear Teeth	**34**		
Sun Gear Teeth	*22*		
Plot pitch diameters?	**n**		
CENTER DISTANCE & PRESSURE ANGLES			
Operating Center Distance		**in**	
Mid-point Center Distance		**in**	
Opr Press Angle - Sun/Planet Mesh		**deg**	
Opr Press Angle - Ring/Planet Mesh		**deg**	
NOMINAL PITCH & PRESSURE ANGLE			
Normal Pitch		**1/in `**	
Normal Pressure Angle		**deg**	

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
Helix Angle	*0.000000*	*deg*	
Transverse Pitch		**1/in `**	
Transverse Press Angle		**deg**	
Axial Pitch		**in**	
Normal Module		**mm `**	
Transverse Module		**mm `**	
PLANET SPACING			
Least mesh angle (Planets must be	**3.2253**	**deg**	
NUMBER OF EQUALLY SPACED PLANETS			
(These are the 1st 4, up to 50, that			
will assemble without interference -			
planets will assemble equally if			
(ring+sun)/planets=integer)			
OPERATION AS GEAR UNIT OR DIFFERENTIAL			
Enable table? 'e=enable 'c=clear	*c*		
Type of unit	**Diff**		
Number of planets			
ck#			
Effective planets (1 member floating)			
Effective planets (All fixed)			
Effective planets			
Planet Interference OD		**in**	
Radial Loads On Sun & Ring Balanced?			

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
Ring/Sun gear ratio	**4.0735**		
Planet/Sun Ratio	**1.5368**		
Ring/Planet Ratio	**2.6507**		
ROTATION SPEED			
Sun gear	*1500.000*	*rpm*	
Ring gear	*-225.000*	*rpm*	
Carrier	*115.000*	*rpm*	
ROTATION SPEED RELATIVE TO CARRIER			
Sun gear	**1385.000**	**rpm**	
Ring gear	**-340.000**	**rpm**	
Planet gear	**-901.244**	**rpm**	
Carrier	**0.000**	**rpm**	
DRIVER/DRIVEN			
Sun gear	**Driver**		
Ring gear	**Driven**		
Carrier	**Driven**		
TORQUES ON ELEMENTS (NO LOSSES)			
Sun gear	*5000.00*	*lbf-in*	
Ring gear	**20367.65**	**lbf-in**	
Carrier	**-25367.65**	**lbf-in**	
POWER (+ IN, - OUT) (NO LOSSES)			
Sun gear	**119.00**	**HP**	
Ring gear	**-72.71**	**HP**	

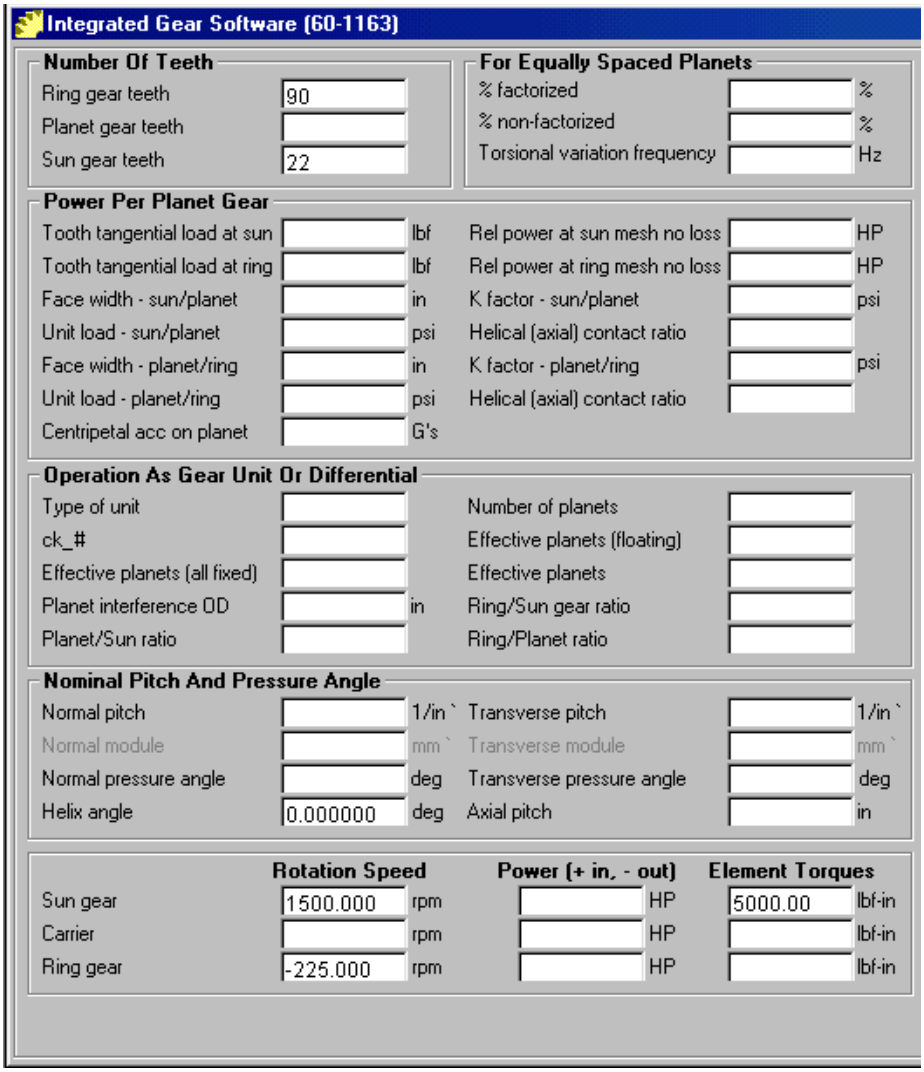
Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
Carrier	**-46.29**	**HP**	
VIBRATION			
Fundamental meshing frequency	**507.83**	**Hz**	
% Factorized		**%**	
% Non-factorized		**%**	
Torsional variation frequency		**Hz**	
APPROXIMATE EFFICIENCY			
Coefficient of Friction	**0.0600**		
Sun/Planet Power Loss		**HP**	
Ring/Planet Power Loss		**HP**	
Total Power Loss (Gear Losses Only)		**HP**	
Approx Efficiency		**%**	

With 22 teeth in the sun and the required speeds we need about 90 teeth in the ring. Enter 90 for the ring gear and blank the carrier speed, as shown in Figure 1-2. After solving again you should have the solution shown in Report 1-2.

Fig. 1-2


Integrated Gear Software (60-1163)
Number Of Teeth
Ring gear teeth
90
Planet gear teeth
Sun gear teeth
22
For Equally Spaced Planets
% factorized
%
% non-factorized
%
Torsional variation frequency
Hz
Power Per Planet Gear
Tooth tangential load at sun
lbf
Tooth tangential load at ring
lbf
Face width - sun/planet
in
Unit load - sun/planet
psi
Face width - planet/ring
in
Unit load - planet/ring
psi
Centripetal acc on planet
G's
Rel power at sun mesh no loss
HP
Rel power at ring mesh no loss
HP
K factor - sun/planet
psi
Helical (axial) contact ratio
K factor - planet/ring
psi
Helical (axial) contact ratio
Operation As Gear Unit Or Differential
Type of unit
ck_#
Effective planets (all fixed)
Planet interference OD
in
Planet/Sun ratio
Number of planets
Effective planets (floating)
Effective planets
Ring/Sun gear ratio
Ring/Planet ratio
Nominal Pitch And Pressure Angle
Normal pitch
1/in `
Normal module
mm `
Normal pressure angle
deg
Helix angle
0.000000
deg
Transverse pitch
1/in `
Transverse module
mm `
Transverse pressure angle
deg
Axial pitch
in
Rotation Speed
Power (+ in, - out)
Element Torques
Sun gear
1500.000
rpm
HP
5000.00
lbf-in
Carrier
rpm
HP
lbf-in
Ring gear
-225.000
rpm
HP
lbf-in

Report 1-2

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
MESSAGE FIELD			
ERROR MESSAGE, internal mesh			
ERROR MESSAGE, external mesh			
ERROR MESSAGE, mesh - general			
Prime factors greater than 100	**none**		
NUMBER OF TEETH			
Ring Gear Teeth	*90*		
Planet Gear Teeth	**34**		
Sun Gear Teeth	*22*		
Plot pitch diameters?	**n**		
CENTER DISTANCE & PRESSURE ANGLES			
Operating Center Distance		**in**	
Mid-point Center Distance		**in**	
Opr Press Angle - Sun/Planet Mesh		**deg**	
Opr Press Angle - Ring/Planet Mesh		**deg**	
NOMINAL PITCH & PRESSURE ANGLE			
Normal Pitch		**1/in `**	
Normal Pressure Angle		**deg**	
Helix Angle	*0.000000*	*deg*	

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
Transverse Pitch		1/in `	
Transverse Press Angle		deg	
Axial Pitch		in	
Normal Module		mm `	
Transverse Module		mm `	
PLANET SPACING			
Least mesh angle (Planets must be	3.2143	deg	
NUMBER OF EQUALLY SPACED PLANETS			
(These are the 1st 4, up to 50, that			
will assemble without interference -			
planets will assemble equally if			
(ring+sun)/planets=integer)			
OPERATION AS GEAR UNIT OR DIFFERENTIAL			
Enable table? 'e=enable 'c=clear	*c*		
Type of unit	Diff		
Number of planets			
ck#			
Effective planets (1 member floating)			
Effective planets (All fixed)			
Effective planets			
Planet Interference OD		in	
Radial Loads On Sun & Ring Balanced?			
Ring/Sun gear ratio	4.0909		

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
Planet/Sun Ratio	**1.5455**		
Ring/Planet Ratio	**2.6471**		
ROTATION SPEED			
Sun gear	*1500.000*	*rpm*	
Ring gear	*-225.000*	*rpm*	
Carrier	**113.839**	**rpm**	
ROTATION SPEED RELATIVE TO CARRIER			
Sun gear	**1386.161**	**rpm**	
Ring gear	**-338.839**	**rpm**	
Planet gear	**-896.928**	**rpm**	
Carrier	**0.000**	**rpm**	
DRIVER/DRIVEN			
Sun gear	**Driver**		
Ring gear	**Driven**		
Carrier	**Driven**		
TORQUES ON ELEMENTS (NO LOSSES)			
Sun gear	*5000.00*	*lbf-in*	
Ring gear	**20454.55**	**lbf-in**	
Carrier	**-25454.55**	**lbf-in**	
POWER (+ IN, - OUT) (NO LOSSES)			
Sun gear	**119.00**	**HP**	
Ring gear	**-73.02**	**HP**	
Carrier	**-45.98**	**HP**	

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
PER PLANET GEAR			
Relative power at sun mesh-no loss		**HP**	
Relative power at ring mesh-no loss		**HP**	
Tooth tangential load at sun		**lbf**	
Tooth tangential load at ring		**lbf**	
Face width - sun/planet		**in**	
K factor - sun/planet		**psi**	
Unit load - sun/planet		**psi**	
Helical (axial) contact ratio			
Face width - planet/ring		**in**	
K factor - planet/ring		**psi**	
Unit load - planet/ring		**psi**	
Helical (axial) contact ratio			
Centripetal acceleration on planet		**G's**	
VIBRATION			
Fundamental meshing frequency	**508.26**	**Hz**	
% Factorized		**%**	
% Non-factorized		**%**	
Torsional variation frequency		**Hz**	
APPROXIMATE EFFICIENCY			
Coefficient of Friction	**0.0600**		
Sun/Planet Power Loss		**HP**	
Ring/Planet Power Loss		**HP**	
Total Power Loss (Gear Losses Only)		**HP**	

With 22 and 90 teeth the carrier speed is about 114 RPM. The sun is a driver and the ring and carrier are driven. The torque and power are listed for all elements. The sun power is positive, indicating power input, and the ring and carrier power are negative, indicating power out. (This should guide any studies of approach and recess action done on the teeth. It is not always apparent. Use the speeds relative to the carrier and relative power for load analysis.)

To size the unit we need to take a guess at the number of planets we will use. Let's assume for now that the number of planets is 4. Enter 4 for "Number of Planets" and solve. The effective number of planets may be different from the actual number of planets, due to errors preventing the planets from sharing the load equally. When one or two members are allowed to float radially the load sharing is better than when all members are constrained by bearings. In this case if floating is utilized the effective number of planets is 3.7. (The effective number of planets will be the same as the actual number only for 3 planets or less with float.) If all members are fixed the effective number of planets is 3. The transmitted torque is divided by the effective number of planets to determine the load for an individual planet. Enter 3.7 for "Effective Planets".

We will use a sun/planet K-factor of 700 for this load condition with about a 1-inch face for now. We want to solve for gear size and normal pitch. To do this will require iteration, so we will toggle to the TK Solver Variable Sheet and enter a "Guess" value of 3 inches for the operating pitch diameter of the sun to give the iterative solver a place to start. (See the *TK Solver User's Manual* for more information about "Guess" values.) Sheet 1-1 shows this entry. If you wish to watch the convergence take place, go to "Settings" under the Options menu and set "Display Intermediate Values?" to Yes. The values will display in the Status Bar as the model is solved. It will slow the program down somewhat.) The solution is shown in Report 1-3.

Sheet 1-1

Variables

Status	Input	Name	Output	Unit	Comment
	1.0000	Fs		in	Face width - sun/planet
	700.00	ks		psi	K factor - sun/planet (See Table)
		uls		psi	Unit load - sun/planet (See Table)
		mhs			Helical (axial) contact ratio
		Fr		in	Face width - planet/ring
		kr		psi	K factor - planet/ring (See Table)
		ulr		psi	Unit load - planet/ring (See Table)
		mhr			Helical (axial) contact ratio
		cf		G's	Centripetal acceleration on planet
					OPERATING PITCH DIAMETERS:
		opr_pdr		in	Ring gear
		opr_pdpr		in	Planet with ring gear
		opr_pdps		in	Planet with sun gear
Guess	3.0000	opr_pds		in	Sun gear

Report 1-3

Input and output report

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
MESSAGE FIELD			
ERROR MESSAGE, internal mesh			
ERROR MESSAGE, external mesh			
ERROR MESSAGE, mesh - general			
Prime factors greater than 100	**none**		

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
NUMBER OF TEETH			
Ring Gear Teeth	*90*		
Planet Gear Teeth	**34**		
Sun Gear Teeth	*22*		
Plot pitch diameters?	**n**		
CENTER DISTANCE & PRESSURE ANGLES			
Operating Center Distance	**3.2095**	**in**	
Mid-point Center Distance	**3.2095**	**in**	
Opr Press Angle - Sun/Planet Mesh		**deg**	
Opr Press Angle - Ring/Planet Mesh		**deg**	
NOMINAL PITCH & PRESSURE ANGLE			
Normal Pitch	**8.724047**	**1/in `**	
Normal Pressure Angle		**deg**	
Helix Angle	*0.000000*	*deg*	
Transverse Pitch	**8.7240**	**1/in `**	
Transverse Press Angle		**deg**	
Axial Pitch		**in**	
Normal Module	**2.911493**	**mm `**	
Transverse Module	**2.9115**	**mm `**	
PLANET SPACING			
Least mesh angle (Planets must be	**3.2143**	**deg**	

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
NUMBER OF EQUALLY SPACED PLANETS			
(These are the 1st 4, up to 50, that	**2**		
will assemble without interference -	**4**		
planets will assemble equally if	#		
(ring+sun)/planets=integer)	#		
OPERATION AS GEAR UNIT OR DIFFERENTIAL			
Enable table? 'e=enable 'c=clear	*c*		
Type of unit	**Diff**		
Number of planets	*4*		
ck#			
Effective planets (1 member floating)	**3.7000**		
Effective planets (All fixed)	**3.0000**		
Effective planets	*3.7000*		
Planet Interference OD	**4.5389**	**in**	
Radial Loads On Sun & Ring Balanced?	**Yes**		
Ring/Sun gear ratio	**4.0909**		
Planet/Sun Ratio	**1.5455**		
Ring/Planet Ratio	**2.6471**		
ROTATION SPEED			
Sun gear	*1500.000*	*rpm*	
Ring gear	*-225.000*	*rpm*	
Carrier	**113.839**	**rpm**	

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
ROTATION SPEED RELATIVE TO CARRIER			
Sun gear	**1386.161**	**rpm**	
Ring gear	**-338.839**	**rpm**	
Planet gear	**-896.928**	**rpm**	
Carrier	**0.000**	**rpm**	
DRIVER/DRIVEN			
Sun gear	**Driver**		
Ring gear	**Driven**		
Carrier	**Driven**		
SUN/PLANET			
Sun/Planet	**915.14**	**ft/min**	
Max Recommended Spacing Tolerance	**0.00115**	**in**	
Min Recommended AGMA Quality Class	**Q8**		
RING/PLANET			
Ring/Planet	**915.14**	**ft/min**	
Max Recommended Spacing Tolerance	**0.00153**	**in**	
Min Recommended AGMA Quality Class	**Q7**		
TORQUES ON ELEMENTS (NO LOSSES)			
Sun gear	*5000.00*	*lbf-in*	
Ring gear	**20454.55**	**lbf-in**	
Carrier	**-25454.55**	**lbf-in**	

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
POWER (+ IN, - OUT) (NO LOSSES)			
Sun gear	**119.00**	**HP**	
Ring gear	**-73.02**	**HP**	
Carrier	**-45.98**	**HP**	
PER PLANET GEAR			
Relative power at sun mesh-no loss	**29.7214**	**HP**	
Relative power at ring mesh-no loss	**-29.7214**	**HP**	
Tooth tangential load at sun	**1071.75**	**lbf**	
Tooth tangential load at ring	**1071.75**	**lbf**	
Face width - sun/planet	*1.0000*	*in*	
K factor - sun/planet	*700.00*	*psi*	
Unit load - sun/planet	**9350.00**	**psi**	
Helical (axial) contact ratio			
Face width - planet/ring		**in**	
K factor - planet/ring		**psi**	
Unit load - planet/ring		**psi**	
Helical (axial) contact ratio			
Centripetal acceleration on planet	**1.181**	**G's**	
OPERATING PITCH DIAMETERS			
Ring gear	**10.3163**	**in**	
Planet with ring gear	**3.8973**	**in**	
Planet with sun gear	**3.8973**	**in**	
Sun gear	**2.5218**	**in**	

Model Title : Program 60-1163

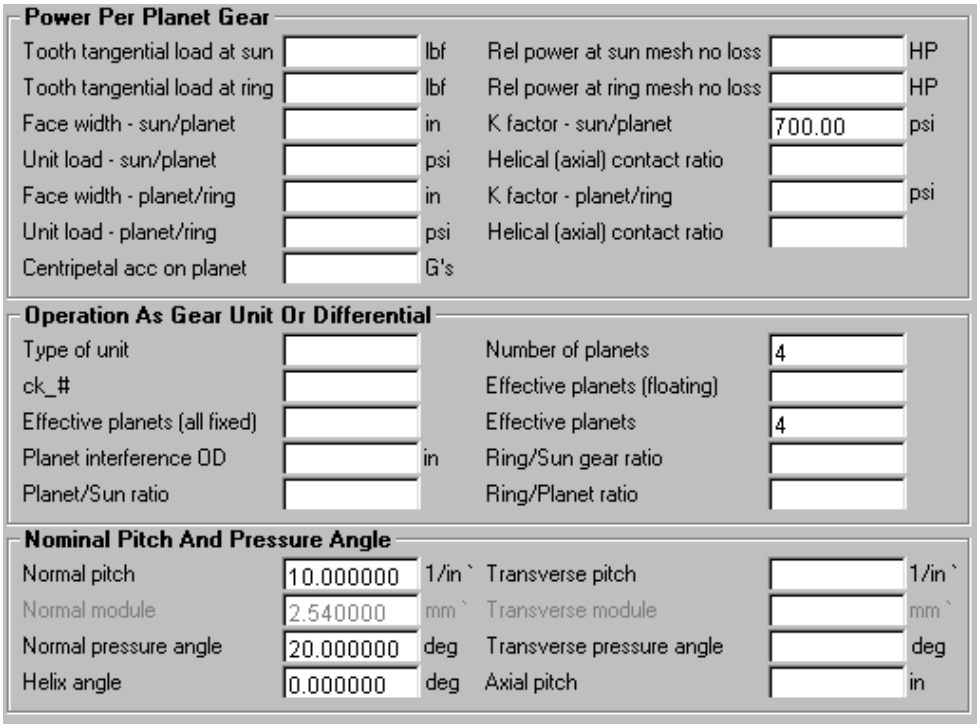
Unit System: US

Description	Value	Unit	Comment
ASPECT RATIOS			
Sun - Face/PD	**0.40**		
Planet - Face/PD (Sun/Planet Mesh)	**0.26**		
VIBRATION			
Fundamental meshing frequency	**508.26**	**Hz**	
% Factorized		**%**	
% Non-factorized		**%**	
Torsional variation frequency		**Hz**	
APPROXIMATE EFFICIENCY			
Coefficient of Friction	**0.0600**		
Sun/Planet Power Loss	**1.10**	**HP**	
Ring/Planet Power Loss	**0.55**	**HP**	
Total Power Loss (Gear Losses Only)	**1.65**	**HP**	
Approx Efficiency	**98.6**	**%**	

The normal pitch for this condition is about 8.72. Four planets is OK for equally spaced assembly. (Note that the effective number of planets from a tooth load standpoint is 3.7. With more than 3 planets, even with one or two members floating, it is very difficult to share the load between planets equally. You may, of course, change the effective number of planets if you desire.) With a 700 K-factor and 1-inch face width the operating pitch diameter of the sun is about 2.5 inches.

Let's suppose we wish to use a little lighter pitch and a higher face to PD ratio. Change the normal pitch to 10 and enter 20 degrees for the normal pressure angle. Blank the face width. The data entry is shown in Figure 1-3, the solved model in Report 1-4.

Fig. 1-3



Power Per Planet Gear

Tooth tangential load at sun		lbf	Rel power at sun mesh no loss		HP
Tooth tangential load at ring		lbf	Rel power at ring mesh no loss		HP
Face width - sun/planet		in	K factor - sun/planet	700.00	psi
Unit load - sun/planet		psi	Helical (axial) contact ratio		
Face width - planet/ring		in	K factor - planet/ring		psi
Unit load - planet/ring		psi	Helical (axial) contact ratio		
Centripetal acc on planet		G's			

Operation As Gear Unit Or Differential

Type of unit			Number of planets	4	
ck_#			Effective planets (floating)		
Effective planets (all fixed)			Effective planets	4	
Planet interference OD		in	Ring/Sun gear ratio		
Planet/Sun ratio			Ring/Planet ratio		

Nominal Pitch And Pressure Angle

Normal pitch	10.000000	1/in `	Transverse pitch		1/in `
Normal module	2.540000	mm `	Transverse module		mm `
Normal pressure angle	20.000000	deg	Transverse pressure angle		deg
Helix angle	0.000000	deg	Axial pitch		in

Report 1-4

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
MESSAGE FIELD			
ERROR MESSAGE, internal mesh	none		
ERROR MESSAGE, external mesh	**none**		
ERROR MESSAGE, mesh - general	**none**		

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
Prime factors greater than 100	**none**		
NUMBER OF TEETH			
Ring Gear Teeth	*90*		
Planet Gear Teeth	**34**		
Sun Gear Teeth	*22*		
Plot pitch diameters?	**n**		
CENTER DISTANCE & PRESSURE ANGLES			
Operating Center Distance	**2.8000**	**in**	
Mid-point Center Distance	**2.8000**	**in**	
Opr Press Angle - Sun/Planet Mesh	**20.0000**	**deg**	
Opr Press Angle - Ring/Planet Mesh	**20.0000**	**deg**	
NOMINAL PITCH & PRESSURE ANGLE			
Normal Pitch	*10.000000*	*1/in `*	
Normal Pressure Angle	*20.000000*	*deg*	
Helix Angle	*0.000000*	*deg*	
Transverse Pitch	**10.0000**	**1/in `**	
Transverse Press Angle	**20.0000**	**deg**	
Axial Pitch		**in**	
Normal Module	**2.540000**	**mm `**	
Transverse Module	**2.5400**	**mm `**	
PLANET SPACING			
Least mesh angle (Planets must be	**3.2143**	**deg**	

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
NUMBER OF EQUALLY SPACED PLANETS			
(These are the 1st 4, up to 50, that	**2**		
will assemble without interference -	**4**		
planets will assemble equally if	#		
(ring+sun)/planets=integer)	#		
OPERATION AS GEAR UNIT OR DIFFERENTIAL			
Enable table? 'e=enable 'c=clear	*'c*		
Type of unit	**Diff**		
Number of planets	*4*		
ck#			
Effective planets (1 member floating)	**3.7000**		
Effective planets (All fixed)	**3.0000**		
Effective planets	*3.7000*		
Planet Interference OD	**3.9598**	**in**	
Radial Loads On Sun & Ring Balanced?	**Yes**		
Ring/Sun gear ratio	**4.0909**		
Planet/Sun Ratio	**1.5455**		
Ring/Planet Ratio	**2.6471**		
ROTATION SPEED			
Sun gear	*1500.000*	*rpm*	
Ring gear	*-225.000*	*rpm*	
Carrier	**113.839**	**rpm**	

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
ROTATION SPEED RELATIVE TO CARRIER			
Sun gear	**1386.161**	**rpm**	
Ring gear	**-338.839**	**rpm**	
Planet gear	**-896.928**	**rpm**	
Carrier	**0.000**	**rpm**	
DRIVER/DRIVEN			
Sun gear	**Driver**		
Ring gear	**Driven**		
Carrier	**Driven**		
SUN/PLANET			
Sun/Planet	**798.37**	**ft/min**	
Max Recommended Spacing Tolerance	**0.00122**	**in**	
Min Recommended AGMA Quality Class	**Q7**		
RING/PLANET			
Ring/Planet	**798.37**	**ft/min**	
Max Recommended Spacing Tolerance	**0.00162**	**in**	
Min Recommended AGMA Quality Class	**Q7**		
TORQUES ON ELEMENTS (NO LOSSES)			
Sun gear	*5000.00*	*lbf-in*	
Ring gear	**20454.55**	**lbf-in**	
Carrier	**-25454.55**	**lbf-in**	

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
POWER (+ IN, - OUT) (NO LOSSES)			
Sun gear	**119.00**	**HP**	
Ring gear	**-73.02**	**HP**	
Carrier	**-45.98**	**HP**	
PER PLANET GEAR			
Relative power at sun mesh-no loss	**29.7214**	**HP**	
Relative power at ring mesh-no loss	**-29.7214**	**HP**	
Tooth tangential load at sun	**1228.50**	**lbf**	
Tooth tangential load at ring	**1228.50**	**lbf**	
Face width - sun/planet	**1.3139**	**in**	
K factor - sun/planet	*700.00*	*psi*	
Unit load - sun/planet	**9350.00**	**psi**	
Helical (axial) contact ratio			
Face width - planet/ring		**in**	
K factor - planet/ring		**psi**	
Unit load - planet/ring		**psi**	
Helical (axial) contact ratio			
Centripetal acceleration on planet	**1.031**	**G's**	
OPERATING PITCH DIAMETERS			
Ring gear	**9.0000**	**in**	
Planet with ring gear	**3.4000**	**in**	
Planet with sun gear	**3.4000**	**in**	
Sun gear	**2.2000**	**in**	

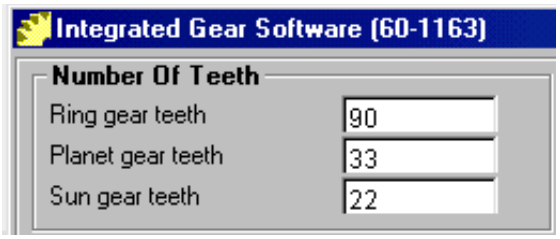
Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
ASPECT RATIOS			
Sun - Face/PD	**0.60**		
Planet - Face/PD (Sun/Planet Mesh)	**0.39**		
VIBRATION			
Fundamental meshing frequency	**508.26**	**Hz**	
% Factorized	**50.0000**	**%**	
% Non-factorized	**50.0000**	**%**	
Torsional variation frequency	**1016.52**	**Hz**	
APPROXIMATE EFFICIENCY			
Coefficient of Friction	**0.0600**		
Sun/Planet Power Loss	**1.10**	**HP**	
Ring/Planet Power Loss	**0.55**	**HP**	
Total Power Loss (Gear Losses Only)	**1.65**	**HP**	
Approx Efficiency	**98.6**	**%**	

With a 34 tooth planet gear and a center distance of 2.8", ϕ_{ext} and ϕ_{int} are both "standard" at 20 degrees. (The "Operating Center Distance" is defaulted to the "Mid-Point Center Distance" if the operating center distance is not entered. If the sun and ring are both odd or both even the mid-point distance will be "standard".) We can assemble 2 or 4 planet gears with equal spacing.

Fig. 1-4



We want ϕ_{ext} to be about 25 degrees, so we need to change the number of planet teeth. (In this case, the condition $N_{ring} = N_{sun} + 2 \cdot N_{planet}$ is met.) Enter 33 for the number of planet teeth. (Changing the planet gear to an odd number of teeth will also double the torsional variation frequency and reduce the amplitude to about half.) Figure 1-4 shows the data entry; Report 1-5 is the solved model.

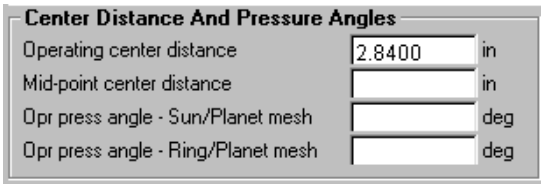
Report 1-5

Model Title : Program 60-1163
Unit System: US

Description	Value	Unit	Comment
MESSAGE FIELD			
ERROR MESSAGE, internal mesh	none		
ERROR MESSAGE, external mesh	**none**		
ERROR MESSAGE, mesh - general	**none**		
Prime factors greater than 100	**none**		
NUMBER OF TEETH			
Ring Gear Teeth	*90*		
Planet Gear Teeth	*33*		
Sun Gear Teeth	*22*		
Plot pitch diameters?	**n**		
CENTER DISTANCE & PRESSURE ANGLES			
Operating Center Distance	**2.8000**	**in**	
Mid-point Center Distance	**2.8000**	**in**	
Opr Press Angle - Sun/Planet Mesh	**22.6444**	**deg**	
Opr Press Angle - Ring/Planet Mesh	**16.9670**	**deg**	

This brings ϕ_{ext} up to about 22.6 degrees and ϕ_{int} down to about 17 degrees. A small change in operating center distance should finish the job. Enter 2.84 for the operating center distance and solve once again (Figure 1-5 and Report 1-6).

Fig. 1-5



Report 1-6

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
MESSAGE FIELD			
ERROR MESSAGE, internal mesh	none		
ERROR MESSAGE, external mesh	**none**		
ERROR MESSAGE, mesh - general	**none**		
Prime factors greater than 100	**none**		
NUMBER OF TEETH			
Ring Gear Teeth	*90*		
Planet Gear Teeth	*33*		
Sun Gear Teeth	*22*		
Plot pitch diameters?	**n**		
CENTER DISTANCE & PRESSURE ANGLES			
Operating Center Distance	*2.8400*	*in*	
Mid-point Center Distance	**2.8000**	**in**	
Opr Press Angle - Sun/Planet Mesh	**24.5066**	**deg**	

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
Opr Press Angle - Ring/Planet Mesh	**19.4381**	**deg**	
NOMINAL PITCH & PRESSURE ANGLE			
Normal Pitch	*10.000000*	*1/in `*	
Normal Pressure Angle	*20.000000*	*deg*	
Helix Angle	*0.000000*	*deg*	
Transverse Pitch	**10.0000**	**1/in `**	
Transverse Press Angle	**20.0000**	**deg**	
Axial Pitch		**in**	
Normal Module	**2.540000**	**mm `**	
Transverse Module	**2.5400**	**mm `**	
PLANET SPACING			
Least mesh angle (Planets must be	**3.2143**	**deg**	
NUMBER OF EQUALLY SPACED PLANETS			
(These are the 1st 4, up to 50, that	**2**		
will assemble without interference -	**4**		
planets will assemble equally if	#		
(ring+sun)/planets=integer)	#		
OPERATION AS GEAR UNIT OR DIFFERENTIAL			
Enable table? 'e=enable 'c=clear	*c*		
Type of unit	**Diff**		
Number of planets	*4*		
ck#			
Effective planets (1 member floating)	**3.7000**		

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
Effective planets (All fixed)	**3.0000**		
Effective planets	*3.7000*		
Planet Interference OD	**4.0164**	**in**	
Radial Loads On Sun & Ring Balanced?	**Yes**		
Ring/Sun gear ratio	**4.0909**		
Planet/Sun Ratio	**1.5000**		
Ring/Planet Ratio	**2.7273**		
ROTATION SPEED			
Sun gear	*1500.000*	*rpm*	
Ring gear	*-225.000*	*rpm*	
Carrier	**113.839**	**rpm**	
ROTATION SPEED RELATIVE TO CARRIER			
Sun gear	**1386.161**	**rpm**	
Ring gear	**-338.839**	**rpm**	
Planet gear	**-924.107**	**rpm**	
Carrier	**0.000**	**rpm**	
DRIVER/DRIVEN			
Sun gear	**Driver**		
Ring gear	**Driven**		
Carrier	**Driven**		

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
SUN/PLANET			
Sun/Planet	**824.50**	**ft/min**	
Max Recommended Spacing Tolerance	**0.00120**	**in**	
Min Recommended AGMA Quality Class	**Q7**		
RING/PLANET			
Ring/Planet	**795.57**	**ft/min**	
Max Recommended Spacing Tolerance	**0.00163**	**in**	
Min Recommended AGMA Quality Class	**Q7**		
TORQUES ON ELEMENTS (NO LOSSES)			
Sun gear	*5000.00*	*lbf-in*	
Ring gear	**20454.55**	**lbf-in**	
Carrier	**-25454.55**	**lbf-in**	
POWER (+ IN, - OUT) (NO LOSSES)			
Sun gear	**119.00**	**HP**	
Ring gear	**-73.02**	**HP**	
Carrier	**-45.98**	**HP**	
PER PLANET GEAR			
Relative power at sun mesh-no loss	**29.7214**	**HP**	
Relative power at ring mesh-no loss	**-29.7214**	**HP**	
Tooth tangential load at sun	**1189.57**	**lbf**	
Tooth tangential load at ring	**1232.83**	**lbf**	

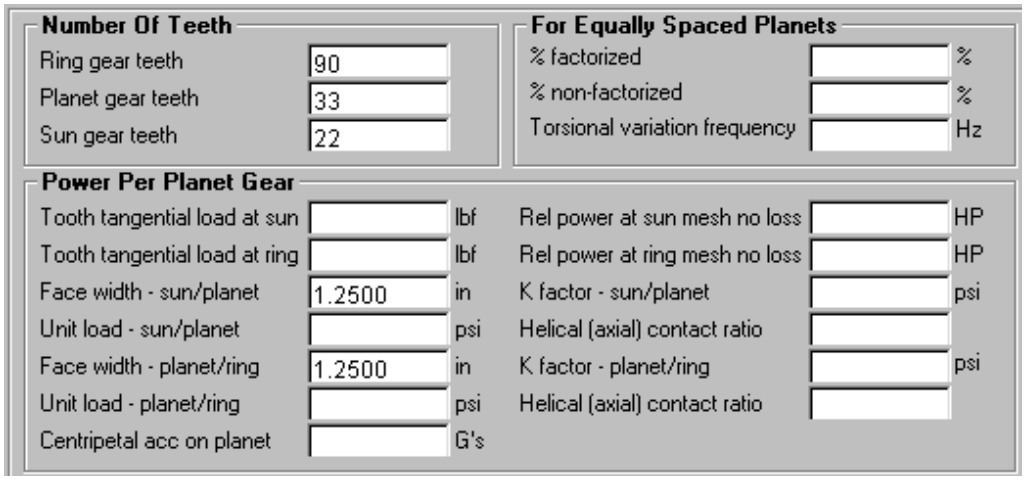
Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
Face width - sun/planet	**1.2466**	**in**	
K factor - sun/planet	*700.00*	*psi*	
Unit load - sun/planet	**9542.40**	**psi**	
Helical (axial) contact ratio			
Face width - planet/ring		**in**	
K factor - planet/ring		**psi**	
Unit load - planet/ring		**psi**	
Helical (axial) contact ratio			
Centripetal acceleration on planet	**1.045**	**G's**	
OPERATING PITCH DIAMETERS			
Ring gear	**8.9684**	**in**	
Planet with ring gear	**3.2884**	**in**	
Planet with sun gear	**3.4080**	**in**	
Sun gear	**2.2720**	**in**	
ASPECT RATIOS			
Sun - Face/PD	**0.55**		
Planet - Face/PD (Sun/Planet Mesh)	**0.37**		
VIBRATION			
Fundamental meshing frequency	**508.26**	**Hz**	
% Factorized	**50.0000**	**%**	
% Non-factorized	**50.0000**	**%**	
Torsional variation frequency	**2033.04**	**Hz**	

We now have ϕ_{ext} at about 24.5 degrees and ϕ_{int} at about 19.4 degrees. The face width for the sun and planet required to keep the K-factor at 700 is 1.247. Let's even the face up to 1 1/4 inch. Blank the 700 K-factor and input 1.25 for the face width of both meshes. (We could use different faces if we wish but, although the compressive stress is much less at the internal mesh, the unit load is about the same, and we should probably set the faces about the same.) The data input is shown in Figure 1-6 and the solved model in Report 1-7.

Fig. 1-6



Report 1-7

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
MESSAGE FIELD			
ERROR MESSAGE, internal mesh	none		
ERROR MESSAGE, external mesh	**none**		
ERROR MESSAGE, mesh - general	**none**		

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
Prime factors greater than 100	**none**		
NUMBER OF TEETH			
Ring Gear Teeth	*90*		
Planet Gear Teeth	*33*		
Sun Gear Teeth	*22*		
Plot pitch diameters?	**n**		
CENTER DISTANCE & PRESSURE ANGLES			
Operating Center Distance	*2.8400*	*in*	
Mid-point Center Distance	**2.8000**	**in**	
Opr Press Angle - Sun/Planet Mesh	**24.5066**	**deg**	
Opr Press Angle - Ring/Planet Mesh	**19.4381**	**deg**	
NOMINAL PITCH & PRESSURE ANGLE			
Normal Pitch	*10.000000*	*1/in `*	
Normal Pressure Angle	*20.000000*	*deg*	
Helix Angle	*0.000000*	*deg*	
Transverse Pitch	**10.0000**	**1/in `**	
Transverse Press Angle	**20.0000**	**deg**	
Axial Pitch		**in**	
Normal Module	**2.540000**	**mm `**	
Transverse Module	**2.5400**	**mm `**	
PLANET SPACING			
Least mesh angle	**3.2143**	**deg**	

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
NUMBER OF EQUALLY SPACED PLANETS			
(These are the 1st 4, up to 50, that	**2**		
will assemble without interference -	**4**		
planets will assemble equally if	#		
(ring+sun)/planets=integer)	#		
OPERATION AS GEAR UNIT OR DIFFERENTIAL			
Enable table? 'e=enable 'c=clear	*c*		
Type of unit	**Diff**		
Number of planets	*4*		
ck#			
Effective planets (1 member floating)	**3.7000**		
Effective planets (All fixed)	**3.0000**		
Effective planets	*3.7000*		
Planet Interference OD	**4.0164**	**in**	
Radial Loads On Sun & Ring Balanced?	**Yes**		
Ring/Sun gear ratio	**4.0909**		
Planet/Sun Ratio	**1.5000**		
Ring/Planet Ratio	**2.7273**		
ROTATION SPEED			
Sun gear	*1500.000*	*rpm*	
Ring gear	*-225.000*	*rpm*	
Carrier	**113.839**	**rpm**	

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
ROTATION SPEED RELATIVE TO CARRIER			
Sun gear	**1386.161**	**rpm**	
Ring gear	**-338.839**	**rpm**	
Planet gear	**-924.107**	**rpm**	
Carrier	**0.000**	**rpm**	
DRIVER/DRIVEN			
Sun gear	**Driver**		
Ring gear	**Driven**		
Carrier	**Driven**		
SUN/PLANET			
Sun/Planet	**824.50**	**ft/min**	
Max Recommended Spacing Tolerance	**0.00120**	**in**	
Min Recommended AGMA Quality Class	**Q7**		
RING/PLANET			
Ring/Planet	**795.57**	**ft/min**	
Max Recommended Spacing Tolerance	**0.00163**	**in**	
Min Recommended AGMA Quality Class	**Q7**		
TORQUES ON ELEMENTS (NO LOSSES)			
Sun gear	*5000.00*	*lbf-in*	
Ring gear	**20454.55**	**lbf-in**	
Carrier	**-25454.55**	**lbf-in**	

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
POWER (+ IN, - OUT) (NO LOSSES)			
Sun gear	**119.00**	**HP**	
Ring gear	**-73.02**	**HP**	
Carrier	**-45.98**	**HP**	
PER PLANET GEAR			
Relative power at sun mesh-no loss	**29.7214**	**HP**	
Relative power at ring mesh-no loss	**-29.7214**	**HP**	
Tooth tangential load at sun	**1189.57**	**lbf**	
Tooth tangential load at ring	**1232.83**	**lbf**	
Face width - sun/planet	*1.2500*	*in*	
K factor - sun/planet	**698.10**	**psi**	
Unit load - sun/planet	**9516.56**	**psi**	
Helical (axial) contact ratio			
Face width - planet/ring	*1.2500*	*in*	
K factor - planet/ring	**189.95**	**psi**	
Unit load - planet/ring	**9862.62**	**psi**	
Helical (axial) contact ratio			
Centripetal acceleration on planet	**1.045**	**G's**	
OPERATING PITCH DIAMETERS			
Ring gear	**8.9684**	**in**	
Planet with ring gear	**3.2884**	**in**	
Planet with sun gear	**3.4080**	**in**	
Sun gear	**2.2720**	**in**	

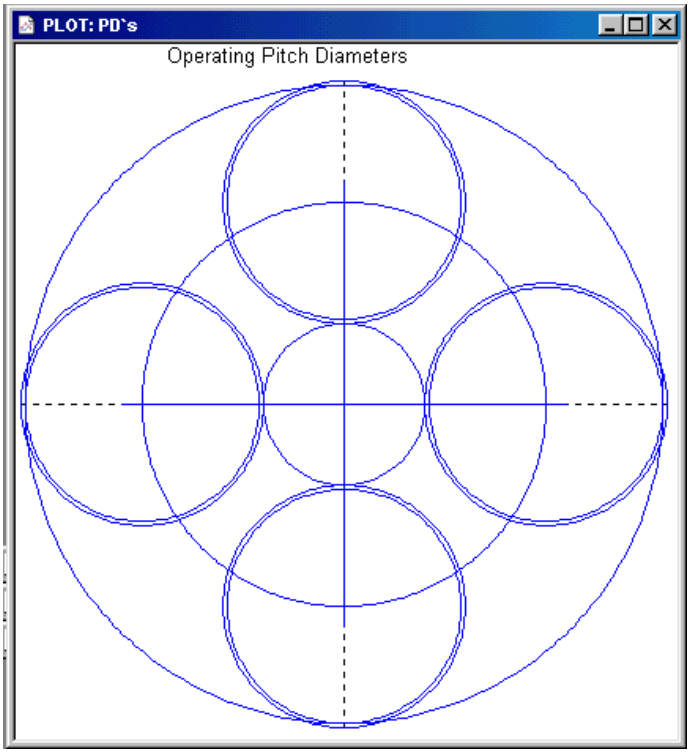
Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
ASPECT RATIOS			
Sun - Face/PD	**0.55**		
Planet - Face/PD (Sun/Planet Mesh)	**0.37**		
VIBRATION			
Fundamental meshing frequency	**508.26**	**Hz**	
% Factorized	**50.0000**	**%**	
% Non-factorized	**50.0000**	**%**	
Torsional variation frequency	**2033.04**	**Hz**	

You can plot operating pitch diameters: check the “Plot pitch diameters?” checkbox and solve. Figure 1-7 is the plot.

Fig. 1-7



The “planets” table in the TK Solver model gives the location of the planet bearings. Open it by selecting it in the Table Sheet or in the drop-down list on the Toolbar. Table 1-1 is the table.

Table 1-1

INT TABLE: planets

Element	Plnt #	Angle, deg	X coord	Y coord	Chord
1	1	0	+2.8400	0	
2					+4.0164
3	2	+90.0000	0	+2.8400	
4					+4.0164
5	3	+180.0000	-2.8400	0	
6					+4.0164
7	4	+270.0000	0	-2.8400	
8					+4.0164
9	1	0	+2.8400	0	

This completes the solution and all design data for the geometry of the epicyclic gear set is solved for in the model. Note that there are no error or caution messages in the error message block. Of course, this is not the only solution to this design problem. The model was solved progressively to obtain this solution. With the multiple-directional solving capability of TK you may wish to investigate other solutions.

This model contains a table with which the relationship among speed, torque and power can be quickly explored. The table uses the geometric data from the model but is independent of the speed, torque and power data.

To use this table, first check off the "Enable table?" checkbox and press Enter. Put the same data as before in the table just to compare results. Your screen should look like Table 1-2.

Table 1-2

INT TABLE: iTable

List	1	2	3	4	5
	Sun	Ring	Carrier	Planet	
Speed	+1500.00	-225.00			
Power					
Torque	+5000.00				
Rel Speed					
Driver/Driven					
PL Velocity					
	Per Planet	Per Planet			
Rel Power					
Tan Load					
K Factor					
Unit Load					
Planet G Load					

Solve and you should have Table 1-3.

Table 1-3

INT TABLE: iTable

List	1	2	3	4	5
	Sun	Ring	Carrier	Planet	
Speed	+1500.00	-225.00	+113.84		
Power	+119.00	-73.02	-45.98		
Torque	+5000.00	+20454.55	-25454.55		
Rel Speed	+1386.16	-338.84	0	-924.11	
Driver/Driven	Driver	Driven	Driven		
PL Velocity	824.50	795.57			
	Per Planet	Per Planet		4	
Rel Power	29.72	-29.72		29.72	
Tan Load	1189.57	1232.83			
K Factor	698.10	189.95			
Unit Load	9516.56	9862.62			
Planet G Load				1.05	

Now let's change the speed and torque in the table. Change your table to match Table 1-4.

Table 1-4

INT TABLE: iTable

List	1	2	3	4
	Sun	Ring	Carrier	Planet
Speed		+150.00		-300.00
Power				
Torque		+2000.00		
Rel Speed	+1386.16	-338.84	0	-924.11
Driver/Driven	Driver	Driven	Driven	
PL Velocity	824.50	795.57		
	Per Planet	Per Planet		4
Rel Power	29.72	-29.72		29.72
Tan Load	1189.57	1232.83		
K Factor	698.10	189.95		
Unit Load	9516.56	9862.62		
Planet G Load				1.05

Solve and you should have Table 1-5.

Table 1-5

INT TABLE: iTable

List	1	2	3	4
	Sun	Ring	Carrier	Planet
Speed	-2140.91	+150.00	-300.00	
Power	-16.61	+4.76	+11.85	
Torque	+488.89	+2000.00	-2488.89	
Rel Speed	-1840.91	+450.00	0	+1227.27
Driver/Driven	Driven	Driver	Driver	
PL Velocity	1094.99	1056.57		
	Per Planet	Per Planet		4
Rel Power	-3.86	3.86		3.86
Tan Load	116.31	120.54		
K Factor	68.26	18.57		
Unit Load	930.51	964.34		
Planet G Load				7.26

Note that with the change of rotation entered we now have both the ring and the carrier driving and the sun as the driven. With the rotation data we originally used we had the sun as the driver and both the ring and carrier were driven. The table makes a note of driver and driven and the algebraic signs of the power for each element also indicates the direction of power flow.

In many situations it can be helpful to investigate the behavior of the elements of a differential over a range of operating conditions. For example, suppose that we wish to use this differential to control the speed of a fan driven by a constant speed electric motor. We will connect the motor to the sun gear, and the carrier to the fan shaft. The motor will also drive a variable speed hydraulic pump-motor combination. The hydraulic unit will also be connected to the ring gear. (This is not meant to imply that this is a good design for this differential unit and is included to illustrate the TK Solver setup for this type of work.) The motor will run at a speed of 1500 RPM. The fan power is known to conform to this equation:

$$\text{Horsepower} = 300(\text{RPM}/500)^3$$

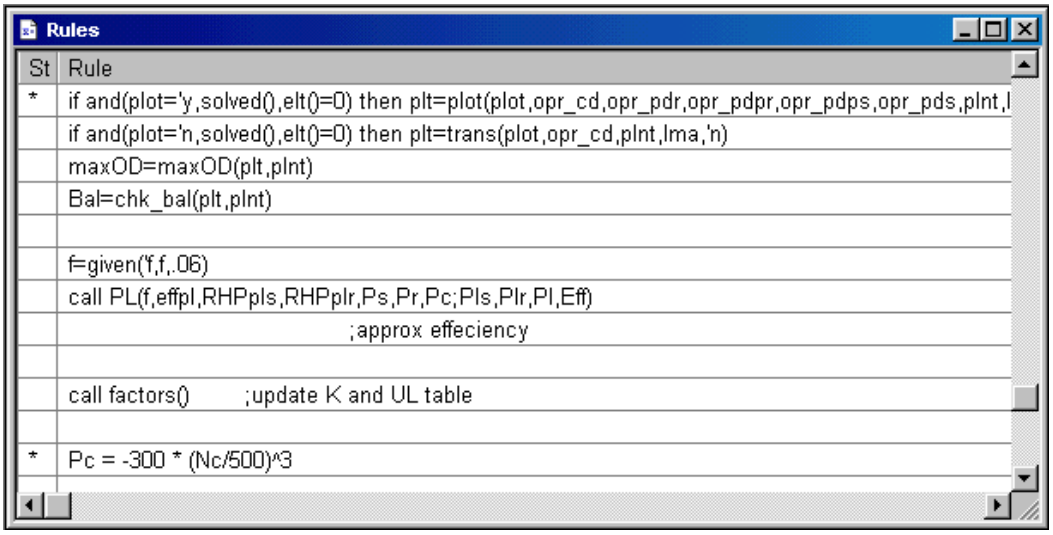
The maximum speed for the fan is 500 RPM where the power is 300 HP. In terms of the variables used in the model this equation would be written:

Pc = -300*(Nc/500)^3

where: Pc = the power removed from the carrier
Nc = the carrier RPM

To set up the model the first step is to enter the equation on the Rule Sheet. Go to the bottom of the Rule Sheet and type in the equation (Sheet 1-2).

Sheet 1-2



Next go to the Variable Sheet and enter 'c for "Enable table?" to avoid waiting for the solution of the table as we proceed. To check the equation we entered on the Rule Sheet, enter 1500 for sun gear speed (electric motor) and 500 for carrier speed (max fan speed). Blank the 5000 for the sun gear torque. The solved model is shown in Report 1-8

Report 1-8

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
MESSAGE FIELD			
ERROR MESSAGE, internal mesh	**none**		
ERROR MESSAGE, external mesh	**none**		
ERROR MESSAGE, mesh - general	**none**		
Prime factors greater than 100	**none**		
NUMBER OF TEETH			
Ring Gear Teeth	*90*		
Planet Gear Teeth	*33*		
Sun Gear Teeth	*22*		
Plot pitch diameters?	**n**		
CENTER DISTANCE & PRESSURE ANGLES			
Operating Center Distance	*2.8400*	*in*	
Mid-point Center Distance	**2.8000**	**in**	
Opr Press Angle - Sun/Planet Mesh	**24.5066**	**deg**	
Opr Press Angle - Ring/Planet Mesh	**19.4381**	**deg**	
NOMINAL PITCH & PRESSURE ANGLE			
Normal Pitch	*10.000000*	*1/in* `	
Normal Pressure Angle	*20.000000*	*deg*	
Helix Angle	*0.000000*	*deg*	

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
Transverse Pitch	**10.0000**	**1/in `**	
Transverse Press Angle	**20.0000**	**deg**	
Axial Pitch		**in**	
Normal Module	**2.540000**	**mm `**	
Transverse Module	**2.5400**	**mm `**	
PLANET SPACING			
Least mesh angle (Planets must be	**3.2143**	**deg**	
NUMBER OF EQUALLY SPACED PLANETS			
(These are the 1st 4, up to 50, that	**2**		
will assemble without interference -	**4**		
planets will assemble equally if	#		
(ring+sun)/planets=integer)	#		
OPERATION AS GEAR UNIT OR DIFFERENTIAL			
Enable table? 'e=enable 'c=clear	*c*		
Type of unit	**Diff**		
Number of planets	*4*		
ck#			
Effective planets (1 member floating)	**3.7000**		
Effective planets (All fixed)	**3.0000**		
Effective planets	*3.7000*		
Planet Interference OD	**4.0164**	**in**	
Radial Loads On Sun & Ring Balanced?	**Yes**		

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
Ring/Sun gear ratio	**4.0909**		
Planet/Sun Ratio	**1.5000**		
Ring/Planet Ratio	**2.7273**		
ROTATION SPEED			
Sun gear	*1500.000*	*rpm*	
Ring gear	**255.556**	**rpm**	
Carrier	*500.000*	*rpm*	
ROTATION SPEED RELATIVE TO CARRIER			
Sun gear	**1000.000**	**rpm**	
Ring gear	**-244.444**	**rpm**	
Planet gear	**-666.667**	**rpm**	
Carrier	**0.000**	**rpm**	
DRIVER/DRIVEN			
Sun gear	**Driver**		
Ring gear	**Driven**		
Carrier	**Driven**		
SUN/PLANET			
Sun/Planet	**594.81**	**ft/min**	
Max Recommended Spacing Tolerance	**0.00138**	**in**	
Min Recommended AGMA Quality Class	**Q7**		
RING/PLANET			
Ring/Planet	**573.94**	**ft/min**	

Model Title : Program 60-1163

Unit System: US

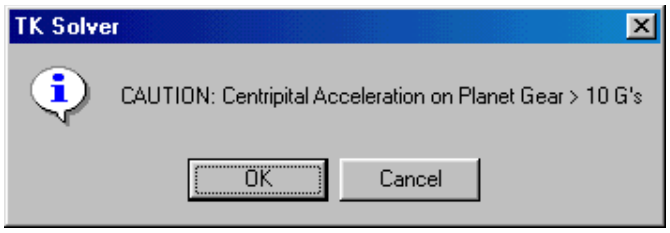
Description	Value	Unit	Comment
Max Recommended Spacing Tolerance	**0.00187**	**in**	
Min Recommended AGMA Quality Class	**Q6**		
TORQUES ON ELEMENTS (NO LOSSES)			
Sun gear	**7427.95**	**lbf-in**	
Ring gear	**30387.05**	**lbf-in**	
Carrier	**-37815.00**	**lbf-in**	
POWER (+ IN, - OUT) (NO LOSSES)			
Sun gear	**176.79**	**HP**	
Ring gear	**123.21**	**HP**	
Carrier	**-300.00**	**HP**	
PER PLANET GEAR			
Relative power at sun mesh-no loss	**31.8533**	**HP**	
Relative power at ring mesh-no loss	**-31.8533**	**HP**	
Tooth tangential load at sun	**1767.21**	**lbf**	
Tooth tangential load at ring	**1831.47**	**lbf**	
Face width - sun/planet	*1.2500*	*in*	
K factor - sun/planet	**1037.10**	**psi**	
Unit load - sun/planet	**14137.70**	**psi**	
Helical (axial) contact ratio			
Face width - planet/ring	*1.2500*	*in*	
K factor - planet/ring	**282.19**	**psi**	
Unit load - planet/ring	**14651.80**	**psi**	

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
Helical (axial) contact ratio			
Centripetal acceleration on planet	**20.166**	**G's**	
OPERATING PITCH DIAMETERS			
Ring gear	**8.9684**	**in**	
Planet with ring gear	**3.2884**	**in**	
Planet with sun gear	**3.4080**	**in**	
Sun gear	**2.2720**	**in**	
ASPECT RATIOS			
Sun - Face/PD	**0.55**		
Planet - Face/PD (Sun/Planet Mesh)	**0.37**		
VIBRATION			
Fundamental meshing frequency	**366.67**	**Hz**	
% Factorized	**50.0000**	**%**	
% Non-factorized	**50.0000**	**%**	
Torsional variation frequency	**1466.67**	**Hz**	
APPROXIMATE EFFICIENCY			
Coefficient of Friction	**0.0600**		
Sun/Planet Power Loss	**1.18**	**HP**	
Ring/Planet Power Loss	**0.59**	**HP**	
Total Power Loss (Gear Losses Only)	**1.77**	**HP**	
Approx Efficiency	**99.4**	**%**	

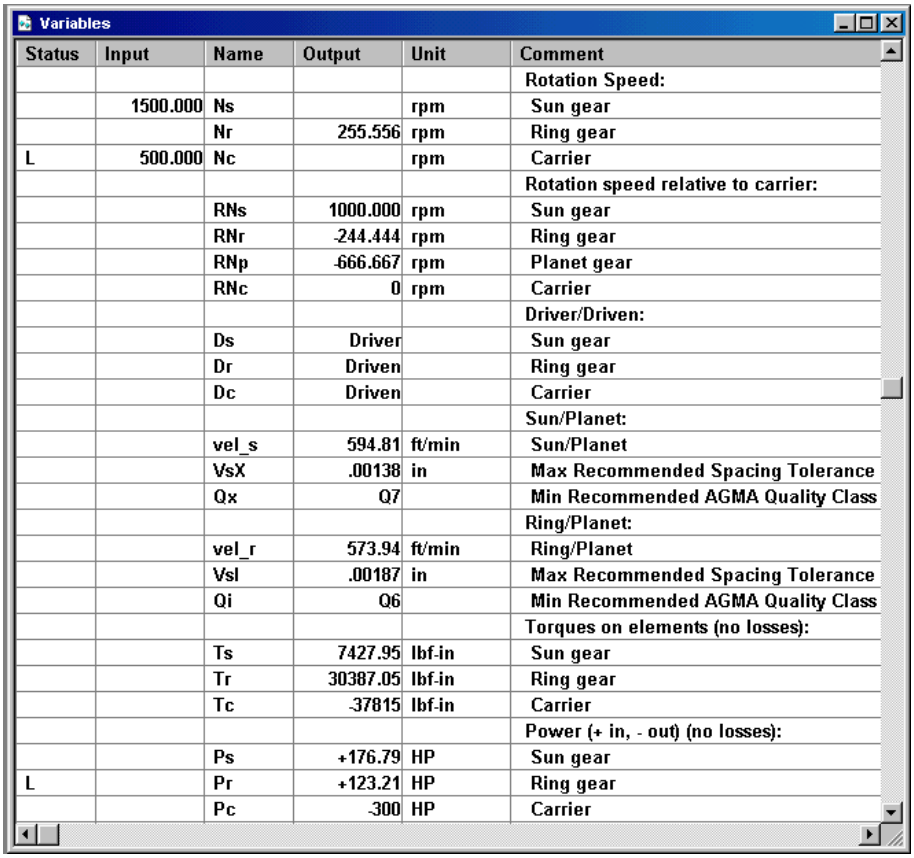
The centripetal acceleration on the planets is quite high, at 20 G's. The model warns you of this:



The planet bearings would have to be capable of running under these conditions. According to the fan speed/power equation, after solving you should have -300 for the carrier horsepower.

Let's check the horsepower carried by the hydraulic system as the fan is operated over a speed range of 0 to 500 RPM. (We could check anything we wish but, for this example, we will check only the hydraulic power against fan speed.) We will use TK Solver's list solving ability to obtain the information we want. The fan speed, which is the same as the carrier speed, varies from 0 to 500 RPM. In the TK Solver Variable Sheet, place the cursor in the status column for carrier speed, Nc, and type "L", or double-click on the cell and pick "List" from the drop-down list that appears.. This will establish Nc as a list. We also need to label the ring gear power, Pr, as a list. (The ring gear power is the same as the hydraulic power as the hydraulic unit is connected to the ring gear.) Your screen should be like Sheet 1-3.

Sheet 1-3



Variables

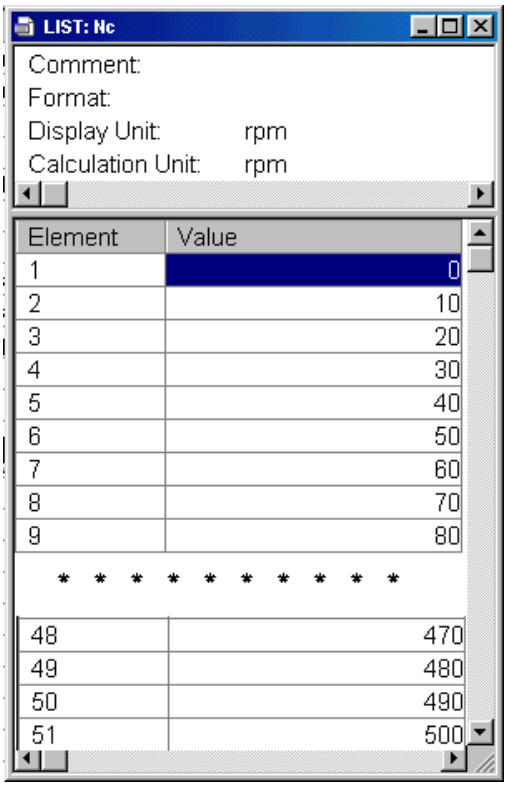
Status	Input	Name	Output	Unit	Comment
					Rotation Speed:
	1500.000	Ns		rpm	Sun gear
		Nr	255.556	rpm	Ring gear
L	500.000	Nc		rpm	Carrier
					Rotation speed relative to carrier:
		RNs	1000.000	rpm	Sun gear
		RNr	-244.444	rpm	Ring gear
		RNp	-666.667	rpm	Planet gear
		RNc	0	rpm	Carrier
					Driver/Driven:
		Ds	Driver		Sun gear
		Dr	Driven		Ring gear
		Dc	Driven		Carrier
					Sun/Planet:
		vel_s	594.81	ft/min	Sun/Planet
		VsX	.00138	in	Max Recommended Spacing Tolerance
		Qx	Q7		Min Recommended AGMA Quality Class
					Ring/Planet:
		vel_r	573.94	ft/min	Ring/Planet
		Vsl	.00187	in	Max Recommended Spacing Tolerance
		Qi	Q6		Min Recommended AGMA Quality Class
					Torques on elements (no losses):
		Ts	7427.95	lbf-in	Sun gear
		Tr	30387.05	lbf-in	Ring gear
		Tc	-37815	lbf-in	Carrier
					Power (+ in, - out) (no losses):
		Ps	+176.79	HP	Sun gear
L		Pr	+123.21	HP	Ring gear
		Pc	-300	HP	Carrier

Both lists have been established but, of course, neither contains any data. We wish Nc, the carrier or fan speed, to be an input list. If there is an entry in the input column then Nc will be an input list. The value on the Variable Sheet will not be used in list solving (unless it is also in the list) but only instructs TK that Nc is an input list. The other list, Pr, the hydraulic power, will be an output list, as there is no entry in the input column.

The next job is to enter the values we desire in the list for Nc. Go to the list subsheet for Nc. (See the TK Solver documentation if necessary.) Instead of typing in the RPM values we need, we will use the automatic list fill feature. Select "Add Step", enter 0 for the first value, 10 for the step size and 500 for the last value. The list will fill with values from 0 to 500 by a step of 10. The screen should look like Sheet 1-4.

Sheet 1-4

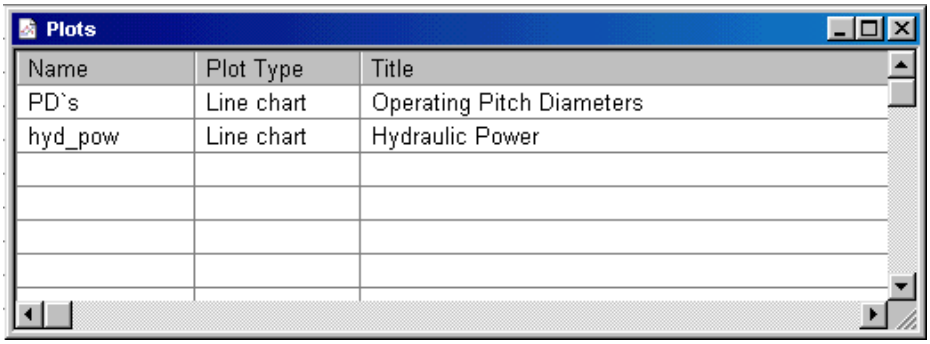
Return to the variable sheet and list solve. The values from the list Nc will be used, one by one, for input. The model will solve and the solution for Pr will be placed in the list for Pr. (We could specify any variables we wish as an output list but we are, for this example, only building a list for Pr.)



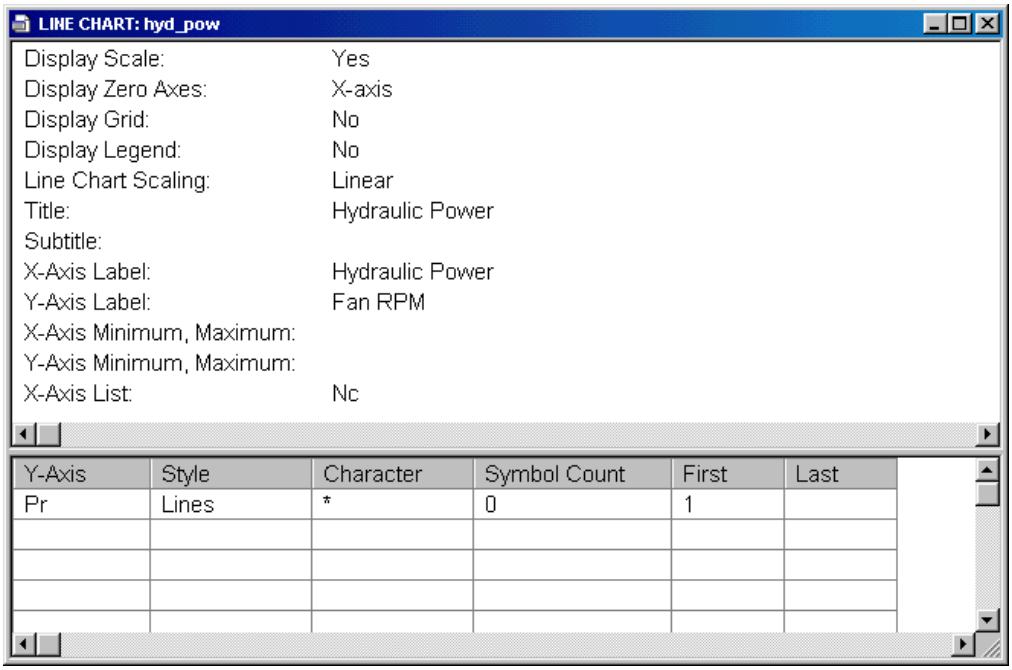
Now that we have a list of fan speeds, Nc, and the corresponding hydraulic power, Pr, we can quickly have a plot of these values. Go to the plot sheet and label a plot "hyd_pow". Select "Line Chart" and type in the title "Hydraulic Power". See Sheet 1-5.

Next, dive into the plot sub-sheet for "hyd_pow," using the right mouse button. Set "Display Zero Axes:" to X-axis, label the X and Y axes, enter Nc as the "X-Axis List" and Pr as the "Y-Axis" list, as shown in Sheet 1-6. Plot (F7 key) and you should have a screen that looks like Figure 1-8.

Sheet 1-5

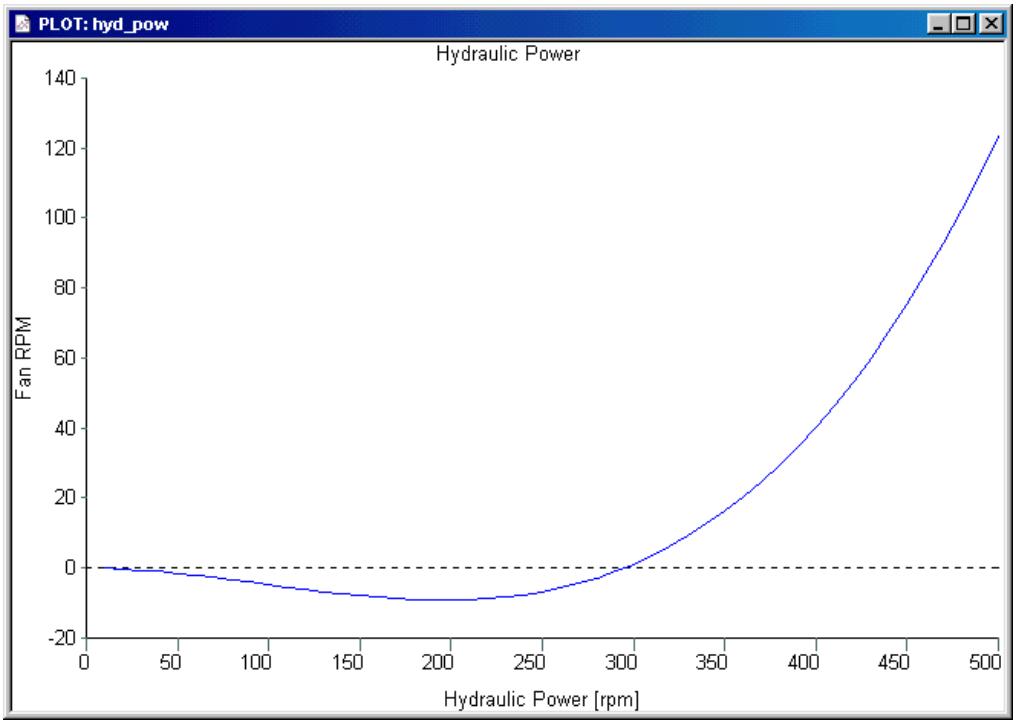


Sheet 1-6



Y-Axis	Style	Character	Symbol Count	First	Last
Pr	Lines	*	0	1	

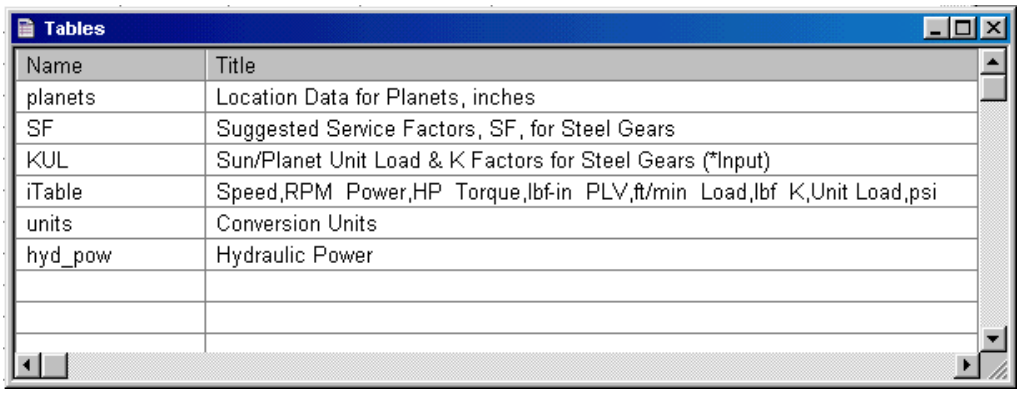
Fig. 1-8



We can see from the plot that power circulates through the differential and back through the hydraulic unit to the motor for fan speeds up to about 290 RPM. At higher fan speeds, power flows from the motor through the hydraulic unit to the differential. The maximum power the hydraulic unit must handle is about 120 HP.

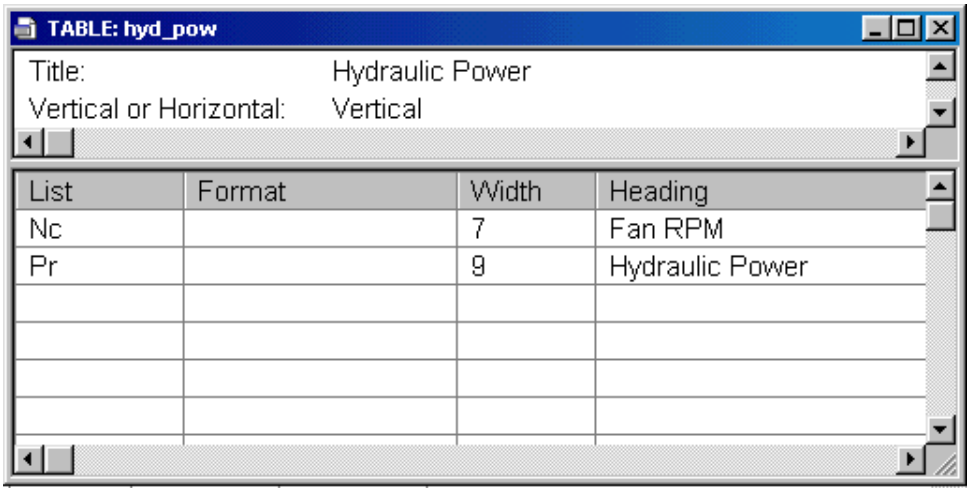
If a table is useful it can be quickly made using the TK Solver Table Sheet. Go to this sheet and label a table "hyd_pow". See Sheet 1-7.

Sheet 1-7



Dive into the table subsheet for "hyd_pow," using the right mouse button, and set up the table (Sheet 1-8).

Sheet 1-8



The table should look like Sheet 1-9.

Sheet 1-9

INT TABLE: hyd_pow

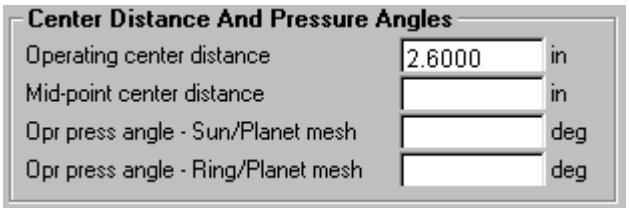
Element	Fan RPM	Hydraulic P
1	0	
2	10	-.0683143
3	20	-.2636571
4	30	-.5716286
5	40	-.9778286
6	50	-1.467857
7	60	-2.027314
8	70	-2.6418
9	80	-3.296914
10	90	-3.978257
11	100	-4.671429
12	110	-5.362029
13	120	-6.035657
14	130	-6.677914
15	140	-7.2744
*	*	*
48	470	92.967343
49	480	102.49509
50	490	112.5726
51	500	123.21429

Note: The relative power in an epicyclic differential is often misunderstood. The input and output torques of any gear unit must balance. The carrier of an epicyclic differential is rotating. Therefore, the meshing velocities of the teeth are different than in a non-epicyclic gear. The power carried by the teeth is a product of load and linear velocity. Since the linear velocity is different than rotation speed multiplied by pitch radius, the relative power is different than the shaft transmitted power. The relative power should be used in load calculations and, of course, the relative speed must then also be used.

Example 2

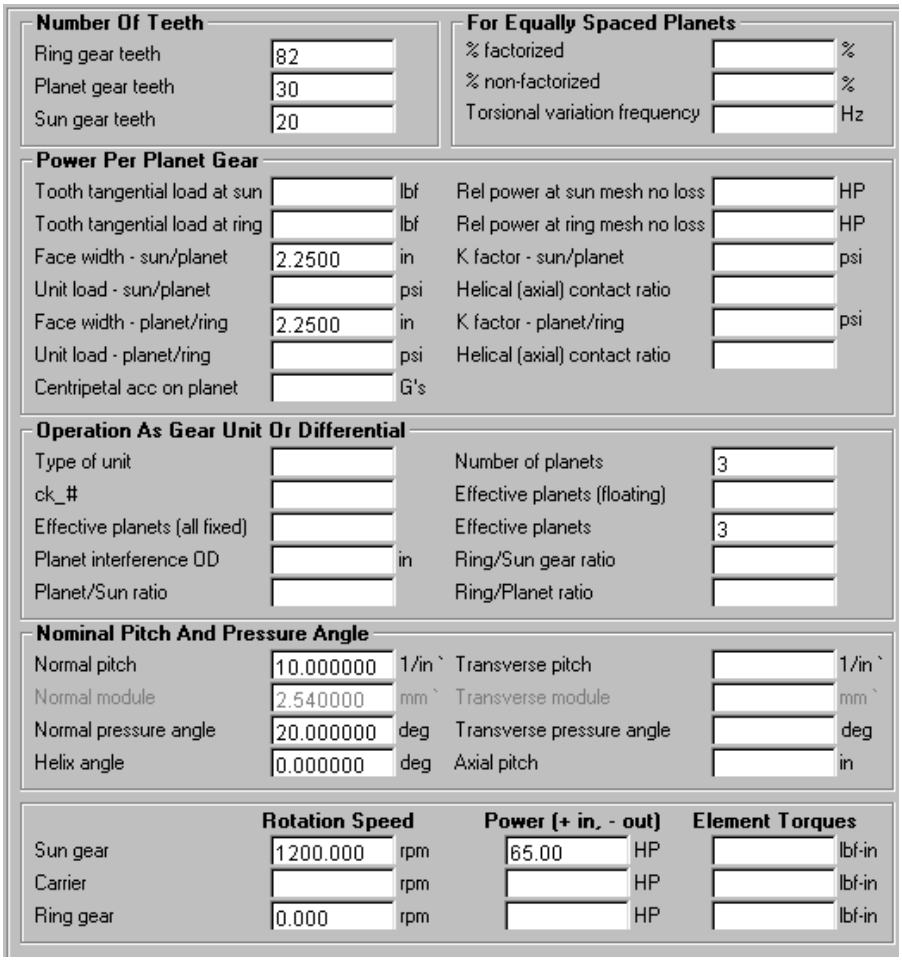
This example is a planetary gear (ring fixed) with a ratio of 5.1 to 1. The ring gear has 82 teeth and the sun gear has 20 teeth. The planet gear has 30 teeth instead of the "standard" 31 teeth in order to bring the operating pressure angle of the external mesh up to about 20 degrees. This is done to increase the beam strength of the external mesh. The operating center distance has been set to "standard" for the internal mesh. (This is not a requirement but this brings the external operating pressure angle to about what we want.)

Fig. 2-1B



The maximum K-factor allowed is 320 psi. However, with three planets the K-factor is about 390 psi which is too high. The sun gear aspect ratio is a little over one and we do not wish to increase it to reduce the K-factor. Figures 2-1A and 2-1B show the data inputs in the wizard form; Report 2-1 shows the solved model.

Fig. 2-1A


Number Of Teeth
Ring gear teeth 82
Planet gear teeth 30
Sun gear teeth 20
For Equally Spaced Planets
% factorized %
% non-factorized %
Torsional variation frequency Hz
Power Per Planet Gear
Tooth tangential load at sun lbf
Tooth tangential load at ring lbf
Face width - sun/planet 2.2500 in
Unit load - sun/planet psi
Face width - planet/ring 2.2500 in
Unit load - planet/ring psi
Centripetal acc on planet G's
Rel power at sun mesh no loss HP
Rel power at ring mesh no loss HP
K factor - sun/planet psi
Helical (axial) contact ratio
K factor - planet/ring psi
Helical (axial) contact ratio
Operation As Gear Unit Or Differential
Type of unit
ck_#
Effective planets (all fixed)
Planet interference OD in
Planet/Sun ratio
Number of planets 3
Effective planets (floating)
Effective planets 3
Ring/Sun gear ratio
Ring/Planet ratio
Nominal Pitch And Pressure Angle
Normal pitch 10.000000 1/in `
Normal module 2.540000 mm `
Normal pressure angle 20.000000 deg
Helix angle 0.000000 deg
Transverse pitch 1/in `
Transverse module mm `
Transverse pressure angle deg
Axial pitch in
Rotation Speed
Power (+ in, - out)
Element Torques
Sun gear 1200.000 rpm 65.00 HP lbf-in
Carrier rpm HP lbf-in
Ring gear 0.000 rpm HP lbf-in

Report 2-1

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
MESSAGE FIELD			
ERROR MESSAGE, internal mesh	none		
ERROR MESSAGE, external mesh	**none**		
ERROR MESSAGE, mesh - general	**none**		
Prime factors greater than 100	**none**		
NUMBER OF TEETH			
Ring Gear Teeth	*82*		
Planet Gear Teeth	*30*		
Sun Gear Teeth	*20*		
Plot pitch diameters?	**n**		
CENTER DISTANCE & PRESSURE ANGLES			
Operating Center Distance	*2.6000*	*in*	
Mid-point Center Distance	**2.5500**	**in**	
Opr Press Angle - Sun/Planet Mesh	**25.3712**	**deg**	
Opr Press Angle - Ring/Planet Mesh	**20.0000**	**deg**	
NOMINAL PITCH & PRESSURE ANGLE			
Normal Pitch	*10.000000*	*1/in `*	
Normal Pressure Angle	*20.000000*	*deg*	
Helix Angle	*0.000000*	*deg*	

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
Transverse Pitch	**10.0000**	**1/in `**	
Transverse Press Angle	**20.0000**	**deg**	
Axial Pitch		**in**	
Normal Module	**2.540000**	**mm `**	
Transverse Module	**2.5400**	**mm `**	
PLANET SPACING			
Least mesh angle (Planets must be	**3.5294**	**deg**	
NUMBER OF EQUALLY SPACED PLANETS			
(These are the 1st 4, up to 50, that	**2**		
will assemble without interference -	**3**		
planets will assemble equally if	#		
(ring+sun)/planets=integer)	#		
OPERATION AS GEAR UNIT OR DIFFERENTIAL			
Enable table? 'e=enable 'c=clear	**e**		
Type of unit	**Planetary**		
Number of planets	*3*		
ck#			
Effective planets (1 member floating)	**3.0000**		
Effective planets (All fixed)	**2.4400**		
Effective planets	*3.0000*		
Planet Interference OD	**4.5033**	**in**	
Radial Loads On Sun & Ring Balanced?	**Yes**		
Ring/Sun gear ratio	**4.1000**		

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
Planet/Sun Ratio	**1.5000**		
Ring/Planet Ratio	**2.7333**		
ROTATION SPEED			
Sun gear	*1200.000*	*rpm*	
Ring gear	*0.000*	*rpm*	
Carrier	**235.294**	**rpm**	
ROTATION SPEED RELATIVE TO CARRIER			
Sun gear	**964.706**	**rpm**	
Ring gear	**-235.294**	**rpm**	
Planet gear	**-643.137**	**rpm**	
Carrier	**0.000**	**rpm**	
DRIVER/DRIVEN			
Sun gear	**Driver**		
Ring gear	**Driven**		
Carrier	**Driven**		
SUN/PLANET			
Sun/Planet	**525.32**	**ft/min**	
Max Recommended Spacing Tolerance	**0.00146**	**in**	
Min Recommended AGMA Quality Class	**Q7**		
RING/PLANET			
Ring/Planet	**505.12**	**ft/min**	
Max Recommended Spacing Tolerance	**0.00198**	**in**	

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
Min Recommended AGMA Quality Class	**Q6**		
TORQUES ON ELEMENTS (NO LOSSES)			
Sun gear	**3413.85**	**lbf-in**	
Ring gear	**13996.80**	**lbf-in**	
Carrier	**-17410.66**	**lbf-in**	
POWER (+ IN, - OUT) (NO LOSSES)			
Sun gear	*65.00*	*HP*	
Ring gear	**0.00**	**HP**	
Carrier	**-65.00**	**HP**	
PER PLANET GEAR			
Relative power at sun mesh-no loss	**17.4183**	**HP**	
Relative power at ring mesh-no loss	**-17.4183**	**HP**	
Tooth tangential load at sun	**1094.18**	**lbf**	
Tooth tangential load at ring	**1137.95**	**lbf**	
Face width - sun/planet	*2.2500*	*in*	
K factor - sun/planet	**389.67**	**psi**	
Unit load - sun/planet	**4863.04**	**psi**	
Helical (axial) contact ratio			
Face width - planet/ring	*2.2500*	*in*	
K factor - planet/ring	**106.91**	**psi**	
Unit load - planet/ring	**5057.56**	**psi**	
Helical (axial) contact ratio			

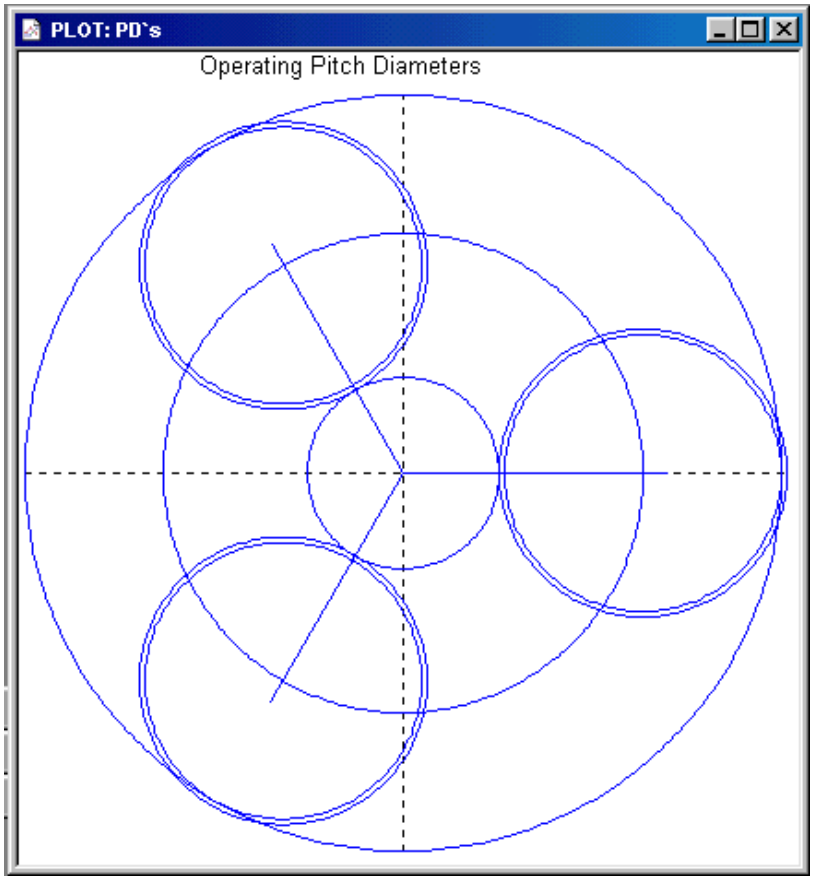
Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
Centripetal acceleration on planet	**4.089**	**G's**	
OPERATING PITCH DIAMETERS			
Ring gear	**8.2000**	**in**	
Planet with ring gear	**3.0000**	**in**	
Planet with sun gear	**3.1200**	**in**	
Sun gear	**2.0800**	**in**	
ASPECT RATIOS			
Sun - Face/PD	**1.08**		
Planet - Face/PD (Sun/Planet Mesh)	**0.72**		
VIBRATION			
Fundamental meshing frequency	**321.57**	**Hz**	
% Factorized	**0.0000**	**%**	
% Non-factorized	**100.0000**	**%**	
Torsional variation frequency	**964.71**	**Hz**	

Figure 2-2 is a plot of the operating pitch diameters.

Fig 2-2



A possible solution to the load problem is to go to 4 planets instead of 3. The sum of the teeth in the ring and sun is 102, which is divisible by 3 but not by 4. Therefore, 4 planets cannot be assembled with equal spacing. However, 102 is divisible by 2, so we may be able to use 2 sets of 2 planets spaced 180 at degrees. Since at least one element is floating we must make sure that the loads on the sun and ring are balanced. We will change to 4 planets and solve the model. After solving we note that the effective number of planets with at least one member floating is 3.7. Enter 3.7 for the effective number of planets and solve again. This solution is shown in Report 2-2.

Report 2-2

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
MESSAGE FIELD			
ERROR MESSAGE, internal mesh	none		
ERROR MESSAGE, external mesh	**none**		
ERROR MESSAGE, mesh - general	**Chk Spc**		
Prime factors greater than 100	**none**		
NUMBER OF TEETH			
Ring Gear Teeth	*82*		
Planet Gear Teeth	*30*		
Sun Gear Teeth	*20*		
Plot pitch diameters?	*y*		
CENTER DISTANCE & PRESSURE ANGLES			
Operating Center Distance	*2.6000*	*in*	

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
Mid-point Center Distance	**2.5500**	**in**	
Opr Press Angle - Sun/Planet Mesh	**25.3712**	**deg**	
Opr Press Angle - Ring/Planet Mesh	**20.0000**	**deg**	
NOMINAL PITCH & PRESSURE ANGLE			
Normal Pitch	*10.000000*	*1/in `*	
Normal Pressure Angle	*20.000000*	*deg*	
Helix Angle	*0.000000*	*deg*	
Transverse Pitch	**10.0000**	**1/in `**	
Transverse Press Angle	**20.0000**	**deg**	
Axial Pitch		**in**	
Normal Module	**2.540000**	**mm `**	
Transverse Module	**2.5400**	**mm `**	
PLANET SPACING			
Least mesh angle (Planets must be	**3.5294**	**deg**	
NUMBER OF EQUALLY SPACED PLANETS			
(These are the 1st 4, up to 50, that	**2**		
will assemble without interference -	**3**		
planets will assemble equally if	**#**		
(ring+sun)/planets=integer)	**#**		
OPERATION AS GEAR UNIT OR DIFFERENTIAL			
Enable table? 'e=enable 'c=clear	**e**		
Type of unit	**Planetary**		
Number of planets	*4*		

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
ck#	**Chk Spc**		
Effective planets (1 member floating)	**3.7000**		
Effective planets (All fixed)	**3.0000**		
Effective planets	*3.7000*		
Planet Interference OD	**3.6199**	**in**	
Radial Loads On Sun & Ring Balanced?	**Yes**		
Ring/Sun gear ratio	**4.1000**		
Planet/Sun Ratio	**1.5000**		
Ring/Planet Ratio	**2.7333**		
ROTATION SPEED			
Sun gear	*1200.000*	*rpm*	
Ring gear	*0.000*	*rpm*	
Carrier	**235.294**	**rpm**	
ROTATION SPEED RELATIVE TO CARRIER			
Sun gear	**964.706**	**rpm**	
Ring gear	**-235.294**	**rpm**	
Planet gear	**-643.137**	**rpm**	
Carrier	**0.000**	**rpm**	
DRIVER/DRIVEN			
Sun gear	**Driver**		
Ring gear	**Driven**		
Carrier	**Driven**		

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
SUN/PLANET			
Sun/Planet	**525.32**	**ft/min**	
Max Recommended Spacing Tolerance	**0.00146**	**in**	
Min Recommended AGMA Quality Class	**Q7**		
RING/PLANET			
Ring/Planet	**505.12**	**ft/min**	
Max Recommended Spacing Tolerance	**0.00198**	**in**	
Min Recommended AGMA Quality Class	**Q6**		
TORQUES ON ELEMENTS (NO LOSSES)			
Sun gear	**3413.85**	**lbf-in**	
Ring gear	**13996.80**	**lbf-in**	
Carrier	**-17410.66**	**lbf-in**	
POWER (+ IN, - OUT) (NO LOSSES)			
Sun gear	*65.00*	*HP*	
Ring gear	**0.00**	**HP**	
Carrier	**-65.00**	**HP**	
PER PLANET GEAR			
Relative power at sun mesh-no loss	**14.1229**	**HP**	
Relative power at ring mesh-no loss	**-14.1229**	**HP**	
Tooth tangential load at sun	**887.18**	**lbf**	
Tooth tangential load at ring	**922.66**	**lbf**	

Model Title : Program 60-1163

Unit System: US

Description	Value	Unit	Comment
Face width - sun/planet	*2.2500*	*in*	
K factor - sun/planet	**315.95**	**psi**	
Unit load - sun/planet	**3943.01**	**psi**	
Helical (axial) contact ratio			
Face width - planet/ring	*2.2500*	*in*	
K factor - planet/ring	**86.68**	**psi**	
Unit load - planet/ring	**4100.73**	**psi**	
Helical (axial) contact ratio			
Centripetal acceleration on planet	**4.089**	**G's**	
OPERATING PITCH DIAMETERS			
Ring gear	**8.2000**	**in**	
Planet with ring gear	**3.0000**	**in**	
Planet with sun gear	**3.1200**	**in**	
Sun gear	**2.0800**	**in**	
ASPECT RATIOS			
Sun - Face/PD	**1.08**		
Planet - Face/PD (Sun/Planet Mesh)	**0.72**		
VIBRATION			
Fundamental meshing frequency	**321.57**	**Hz**	
% Factorized		**%**	
% Non-factorized		**%**	
Torsional variation frequency		**Hz**	

This brings the K-factor for the sun and planet mesh down to about 316 psi, which is less than the maximum of 320 psi. The loads on the sun and ring are balanced with this arrangement. The TK Solver table "planets" will give us the proper location for the planets to make assembly possible and insure that the sun and ring loads are balanced. See Sheet 2.

Sheet 2

INT TABLE: planets

Element	Plnt #	Angle, deg	X coord	Y coord	Chord
1	1	0	+2.6000	0	
2					+3.6199
3	2	+88.2353	+0.0801	+2.5988	
4					+3.7331
5	3	+180.0000	-2.6000	0	
6					+3.6199
7	4	+268.2353	-0.0801	-2.5988	
8					+3.7331
9	1	0	+2.6000	0	

Figure 2-3 is a plot of the operating pitch diameters. Note that the planets are in pairs 180 degrees apart but the pairs are not at 90 degrees.

Fig 2-3